Engineering Case Library

ECL 115A

Design of a Scotch Yoke Mechanism

Summary

For six weeks the Scotch Yoke mechanism had operated beautifully. It had been giving a long, slender, induction coil a smooth, gentle oscillating motion of $\pm 110^{\circ}$ in a magnetic field at a rate of three cycles per minute–continuously. The output of the coil was repetitive within one part per 10^4. Then (it seemed to happen very suddenly) the output readings began to be erratic and an examination of the Scotch Yoke indicated that it was chattering and "hanging up" momentarily at one point in its cycle and then "catching up" with a sudden rush. This malfunction caused some acute embarrassment to Aaron Baumgarten, the engineer responsible for the design of the Scotch Yoke. He had been asked to design a replacement drive mechanism for an existing drive which was unsatisfactory. His reaction to the problem had been that it was not an especially demanding one and that it could be readily solved–and he had said so. For this reason, the malfunction of the Scotch Yoke gave Aaron's colleagues a wonderful opportunity to pull his leg--and they did not waste the opportunity. He quickly reacted with corrective action which should have been adequate but fate, in the form of a technician (working on the night shift) stepped in to confound him and another malfunction occurred making corrective action necessary once again. This time all went well and the unit performed satisfactorily for four months.

 Prepared by Mr. Aaron Baumgarten with support from the National Science Foundation through the Case Program, Design Division, Department of Mechanical Engineering.

Background

The long, slender, induction coil (Figure 1) is one of the elements in the control system for the electron beam at the Stanford Linear Accelerator Center. The voltage induced in this coil constitutes an input to a computer which acts to maintain the conditions prescribed for a specific experiment. The maintenance of these conditions is important to the experimental physicists because they represent a known datum which is used in the interpretation of the data accumulated during the experiment.

The induction coil is essentially a plastic rod about one inch square and about seventeen feet long on which a longitudinal coil has been formed. The plastic rod had been formed by bonding plastic strips into a laminated rod having the desired cross-section. Plastic wheels which were bonded to the rod at regular intervals acted to support the rod in "bogey" wheels which were carried in a supporting cradle. The entire ensemble was then inserted between the pole pieces of a large di-pole electro magnet which was performing a gaging function by indicating the field strength of other magnets located in the actual electron beam. The coil was long enough to protrude from both ends of the magnet, thereby providing a place to attach some driving means. One attempt had been made to use limit switches to periodically reverse the direction of motion of a drive motor coupled to the coil (see Exhibit A-1). This scheme was judged to be unsatisfactory because of its erratic behavior. The specific cause of this undesirable behavior was never firmly established. Some of the people involved thought that the sudden acceleration imparted to the coil with each reversal of motion might be twisting the coil so that the end which was closest to the driving motor was out of phase with the end furthest from the motor. Others felt that the magnetic clutch in the drive might be slipping erratically. Everyone involved agreed–there was a problem to be solved. After some discussion it was decided that a new approach might be a good way to eliminate the problem and the following specifications for a new drive were agreed upon:

a. The coil was to be oscillated in such a way as to impart a gentle acceleration to it.

b. The drive means were to be highly reliable. (As one man put it, "We want it to run forever with zero maintenance.") A design goal of 10,000 hours of operation was agreed upon.

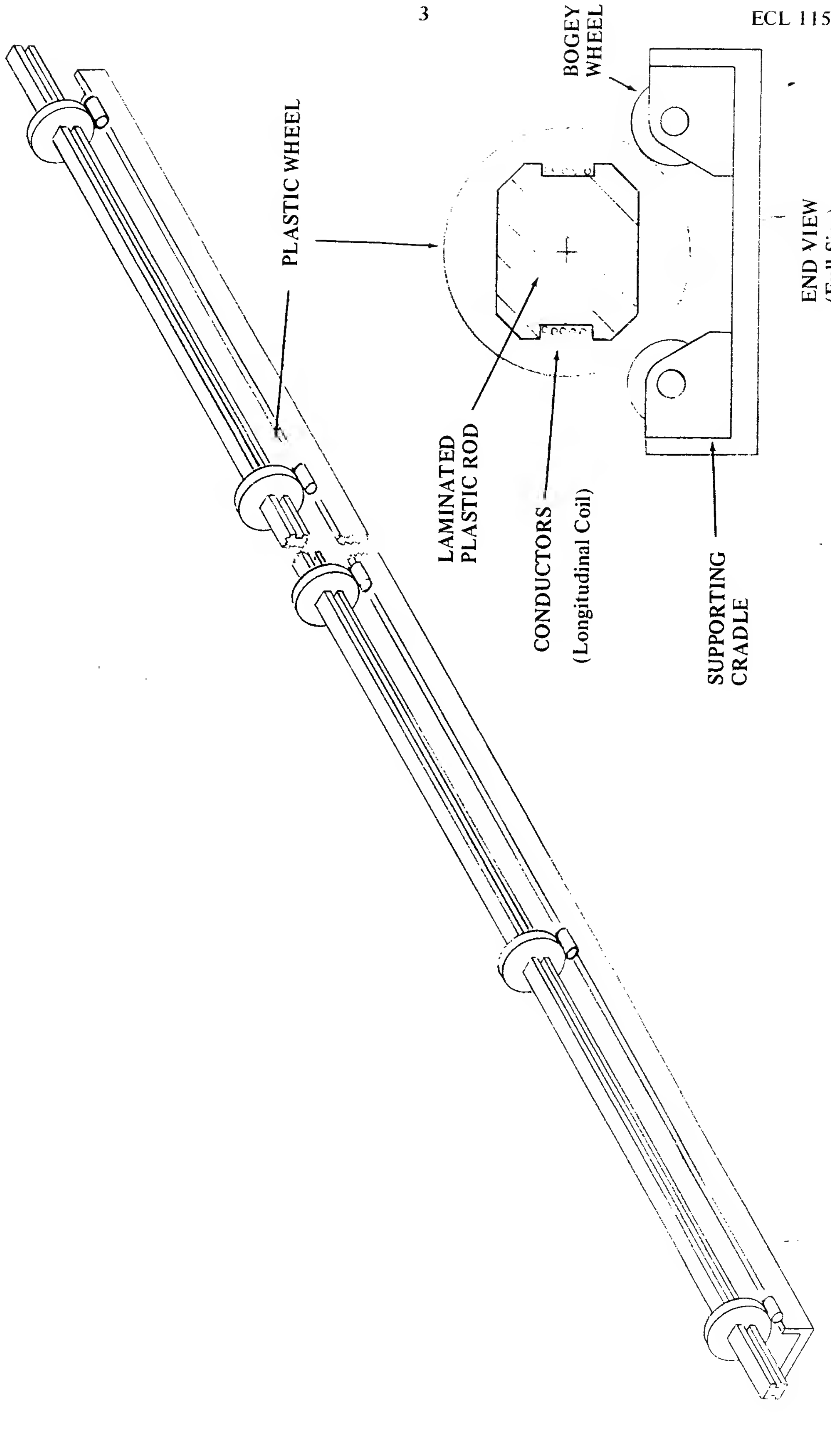
c. The cost of materials and shop fabrication was not to exceed $500.

First Scotch Yoke Design

Aaron suggested that kinematically, a sinusoidal (i.e., simple harmonic) motion would be sufficiently gentle and that mechanisms for generating such motion were well known and easy to build. For example, an electric motor driving any of the following devices (Figure 2) seemed well suited for the job.

a. A simple eccentric
b. A Hypocycloid
c. A Swash Plate
d. An Isosceles Linkage*
e. A Slider Crank Mechanism (which approximates a sinusoidal output)
f. A Scotch Yoke Mechanism

*See: *Elements of Mechanism*, Doughtie & James. J. Wiley



Sketch of Coil

Figure 1.

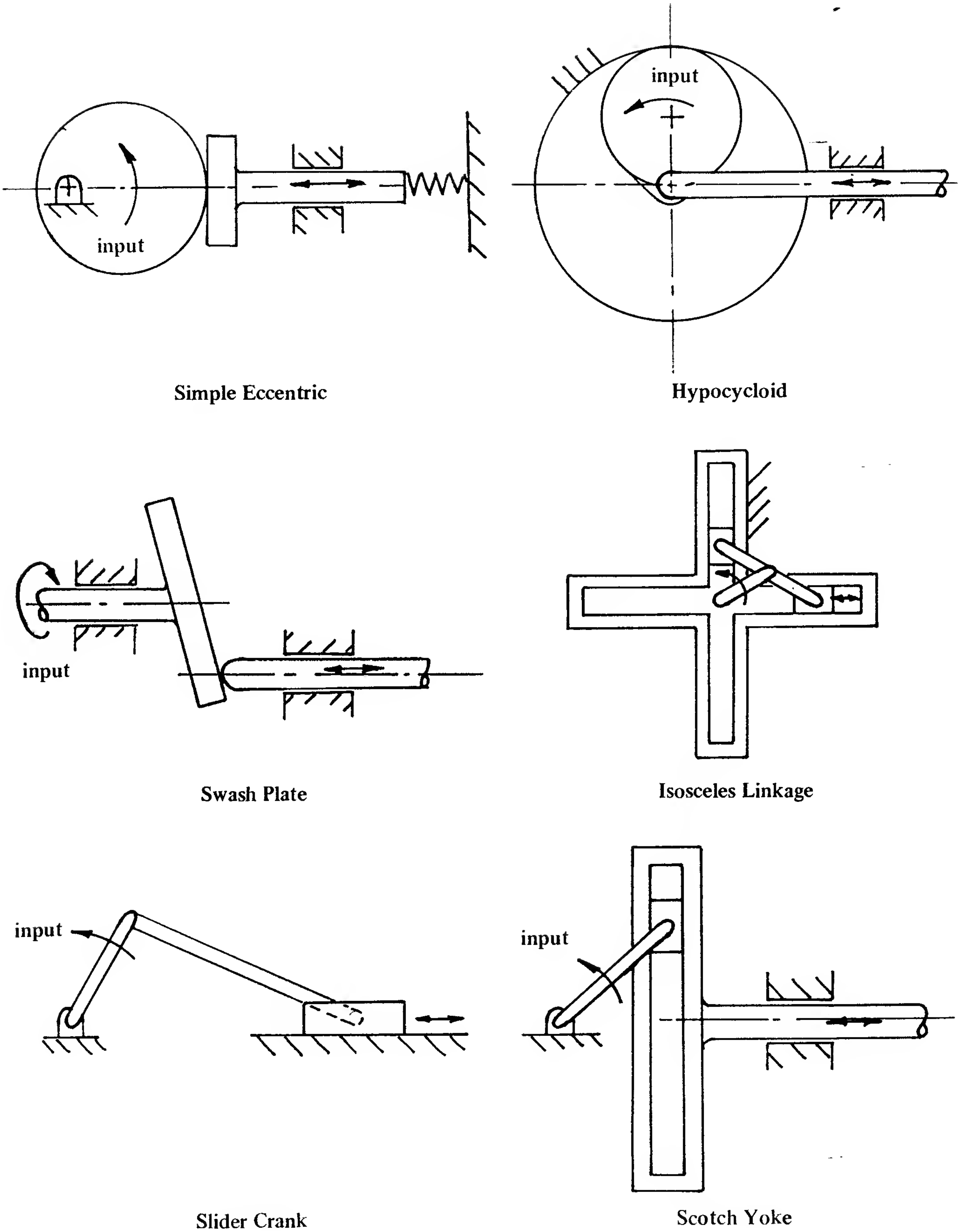


Figure 2

Sketches of Alternative Mechanisms

There were other devices which were kinematically superior in terms of their acceleration characteristics. However, the estimated cost of manufacture in every case was higher.

The initial design concepts (Figure 3) considered the use of an eccentrically mounted circular disc cam. Such a cam would impart simple harmonic motion to a reciprocating follower which in turn could then drive the induction coil in a rotary oscillatory fashion through a rack and pinion. This was an extremely attractive approach because each of the components was either easy to machine or could be readily purchased. However, after considering several ways in which to make this drive positive (i.e., spring loaded, gravity loaded, face type groove, double acting follower) (Figure 3), Aaron felt that the Scotch Yoke configuration (Figure 4) was a more desirable one.

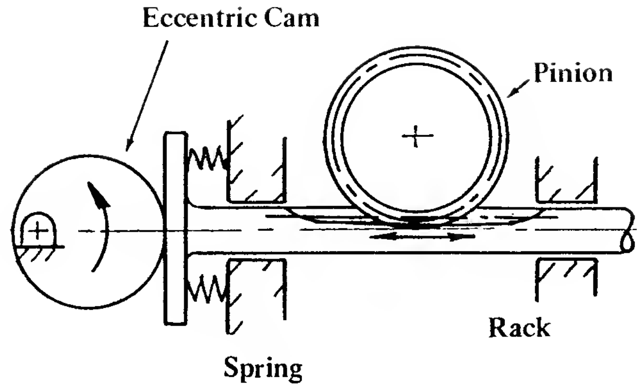
The decision was made and a dynamic analysis of the mechanism was done (Appendix A-1). This analysis indicated that the loads, friction forces and deflections which would result from normal operation were very small. On this basis, the design was completed (with much attention to cost of fabrication) and the drawings were released. (Exhibits A-2a through A-2c show some representative details)

Fabrication of the unit was uneventful and was completed quickly (see Exhibit A-3). The original drive motor (Bodine Type K-2; Frame Type KYC-22 RM Cat. B8122E-600M – Exhibit A-4) and the original coil were coupled together through the Scotch Yoke mechanism. Initial operation was very good and after some weeks of smooth, trouble-free operation, Aaron considered the case "closed".

Failure of First Scotch Yoke

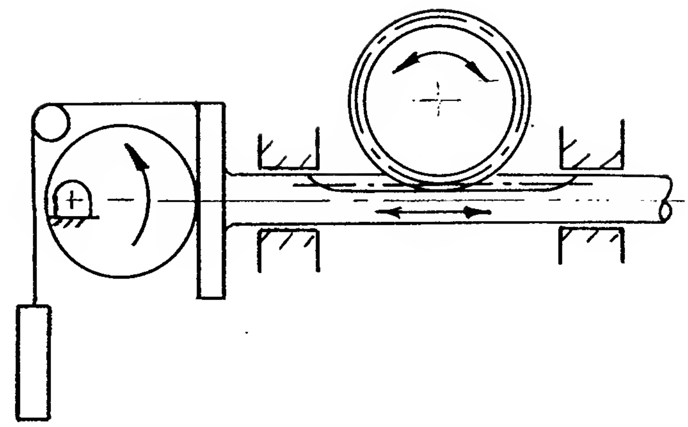
This comfortable frame of mind lasted for about six weeks until he received a phone call which sounded like, "Hey, your coil-flipping mechanism isn't working right. You'd better get down here and fix it right away." A visual examination disclosed the fact that the horizontal rack (see Exhibit A-3) had two scratch-like marks high on its periphery evidently created by the two rows of balls in the anti-friction ball bushings which supported the rack. In addition to this, the vertical, hardened steel shaft in the yoke showed a series of horizontal light and dark rings along its central portion. The conclusion drawn from this evidence was that the vertical slider running on this shaft was chattering, and careful observation (visual and tactile) supported this conclusion. The chattering was so slight in magnitude that an observer looking at the bushing housing would suspect that chattering was there but couldn't be sure. However, placing the fingertips lightly on the bushing housing enabled an observer to feel, quite distinctly, a vibration which occurred as the slider moved up along its shaft.

The explanation of this situation which seemed most reasonable to Aaron was as follows: The friction force exerted by the slider on its shaft was reaching a value greater than he had originally estimated. Further, the highest friction force was acting when the rack presented its greatest unsupported length. It was therefore being deflected more than the calculated value. This in turn was creating a very high local load between the balls of



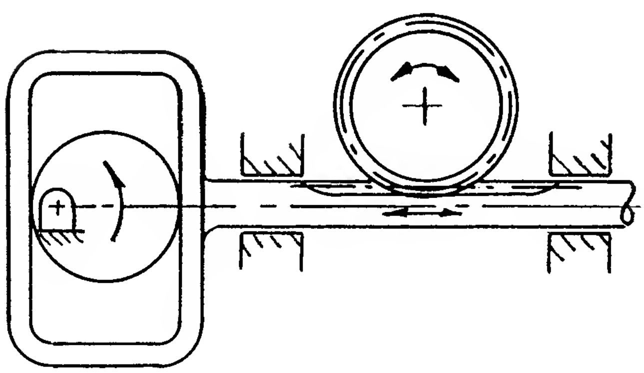
Spring Loaded

Eccentric Cam



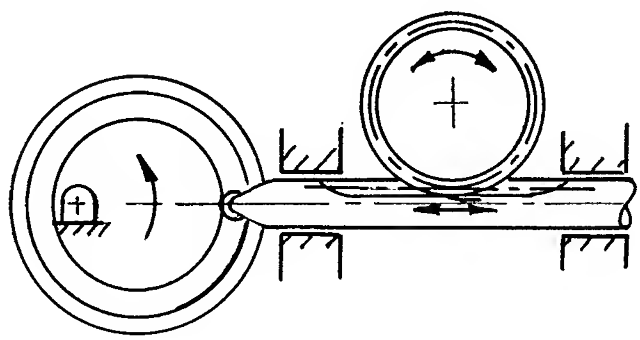
Gravity Loaded

Eccentric Cam



Double Acting Follower

Eccentric Cam



Face Type Groove

Eccentric Cam

Figure 3

Sketch of Circular Cam

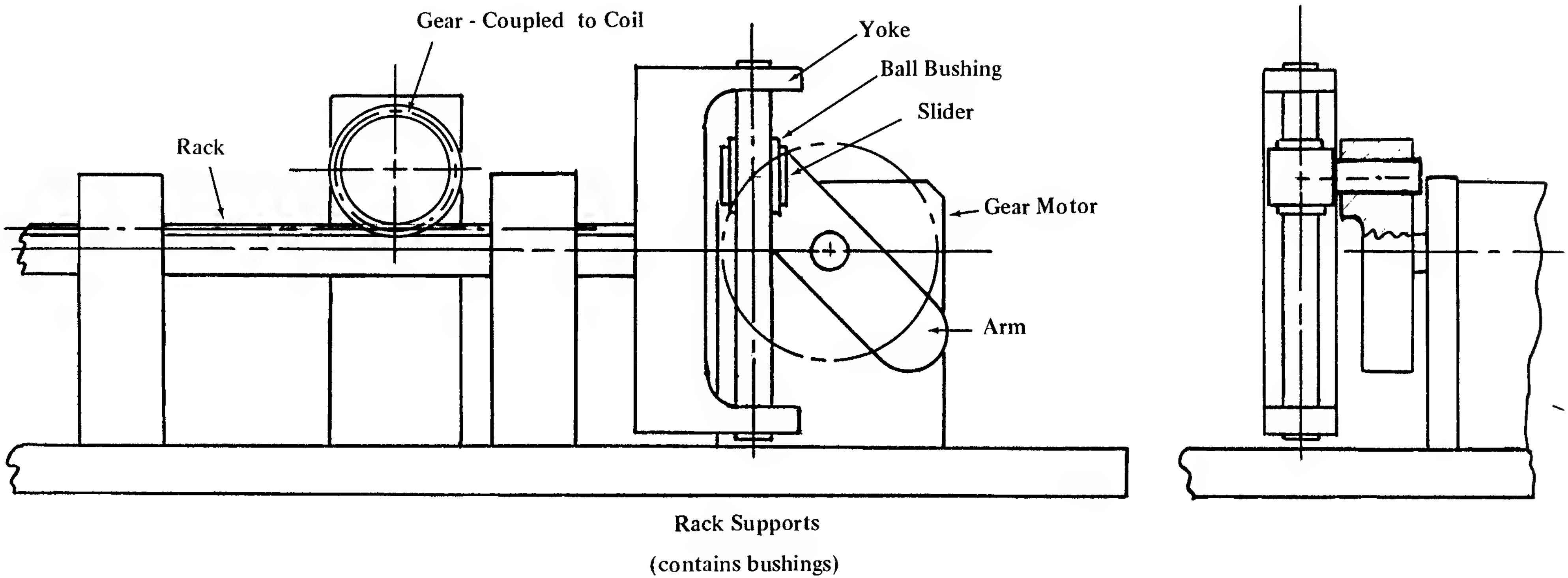


Figure 4

Schematic of First Scotch Yoke

the ball bushing and the "soft" rack (i.e., 303 Stainless Steel). This high local loading was responsible for the scratch-like marks observed on the rack. Once this action was started, it became progressively worse until enough free play was generated to cause the mechanism to malfunction. (Here, a malfunction means a rough oscillation of the induction coil which was reflected in nonrepeatable output readings from the computer.)

To "fix it right away" Aaron had the mechanism brought back to the shop where an additional constraint was built into the framework in such a manner as to resist any upward deflection of the rack in its extended position (see Exhibit A-5). No such constraint was considered necessary for downward deflection because the unsupported length of rack was very short when it experienced a downward force. In addition to this added constraint, the ball bushings were removed and replaced with nylon bushings.

The unit was operated for several hours and ran smoothly – therefore it was put back into service where it performed well for several days. Over the weekend, however, one of the technicians on a night shift decided to help the mechanism along by liberally oiling the bearings. On the following Monday morning Aaron noticed this fact and removed as much of the oil as he could -- then told the supervising technician to instruct his people not to oil the device anymore. He took this action because he knew that nylon, unless it was in a moisture stabilized form, would absorb water and swell. In this case the nylon bushing was not moisture stabilized and Aaron assumed that swelling could be caused by absorption of oil as well as by water.

His verbal request didn't prove to be an effective way to communicate, however, because on Tuesday morning the unit was again found to be liberally covered with oil and by now the nylon bushings were grabbing their respective shafts so strongly that the operation of the mechanism was visibly jumpy. Aaron then decided to go back to a ball bushing on the vertical slider and to replace the nylon bushings supporting the horizontal rack with similar bushings which had larger holes. The reasoning behind this move was that the horizontal bushings could have a very liberal clearance without seriously affecting the performance of the mechanism. With a liberal clearance it was possible to believe that even a recurrence of the "oiling" procedure could be tolerated without adverse affects on the performance.

The vertical slider presented a different problem – in order to prevent wobbling of the slider a limited clearance between bushing and shaft was felt to be necessary. The alternative was to make the slider longer – but time did not permit. One more thing was done: A sign was put in the reinstalled unit asking one and all to refrain from oiling the mechanism. These actions seemed to do the job, because the unit performed well.

Second Scotch Yoke

Sometime later the need for a second drive unit arose. Aaron decided to change its form in an attempt to avoid some of the faults of the first design. This meant abandoning the price ceiling which influenced the design of the first unit, but in light of his experience this was considered to be a worthwhile thing to do

and a redesign was started. This time a more carefully constrained mechanism was developed, and the rack was divorced from any contact with bushings. In order to eliminate any lingering doubts about maintaining geometric fidelity, the unit was made quite massive, (see Exhibits A-6a through A-6c) following in a sense, the philosophy of machine tool builders who traditionally achieve rigidity by making their machines massive. One more thing was done to eliminate a theoretically negligible force which might have been gremlin-like in its nature. Whereas the first Scotch Yoke had actuated two micro-switches (one each at the end of a half-cycle), the second design accomplished this switching function by the interruption of a beam of light focussed on a photo-diode at the proper time (see Exhibit A-7). This second drive remained in continuous, trouble-free operation for three months (except for the fact that the bulbs acting on the photo-diode switches needed changing one time.)

LIST OF EXHIBITS

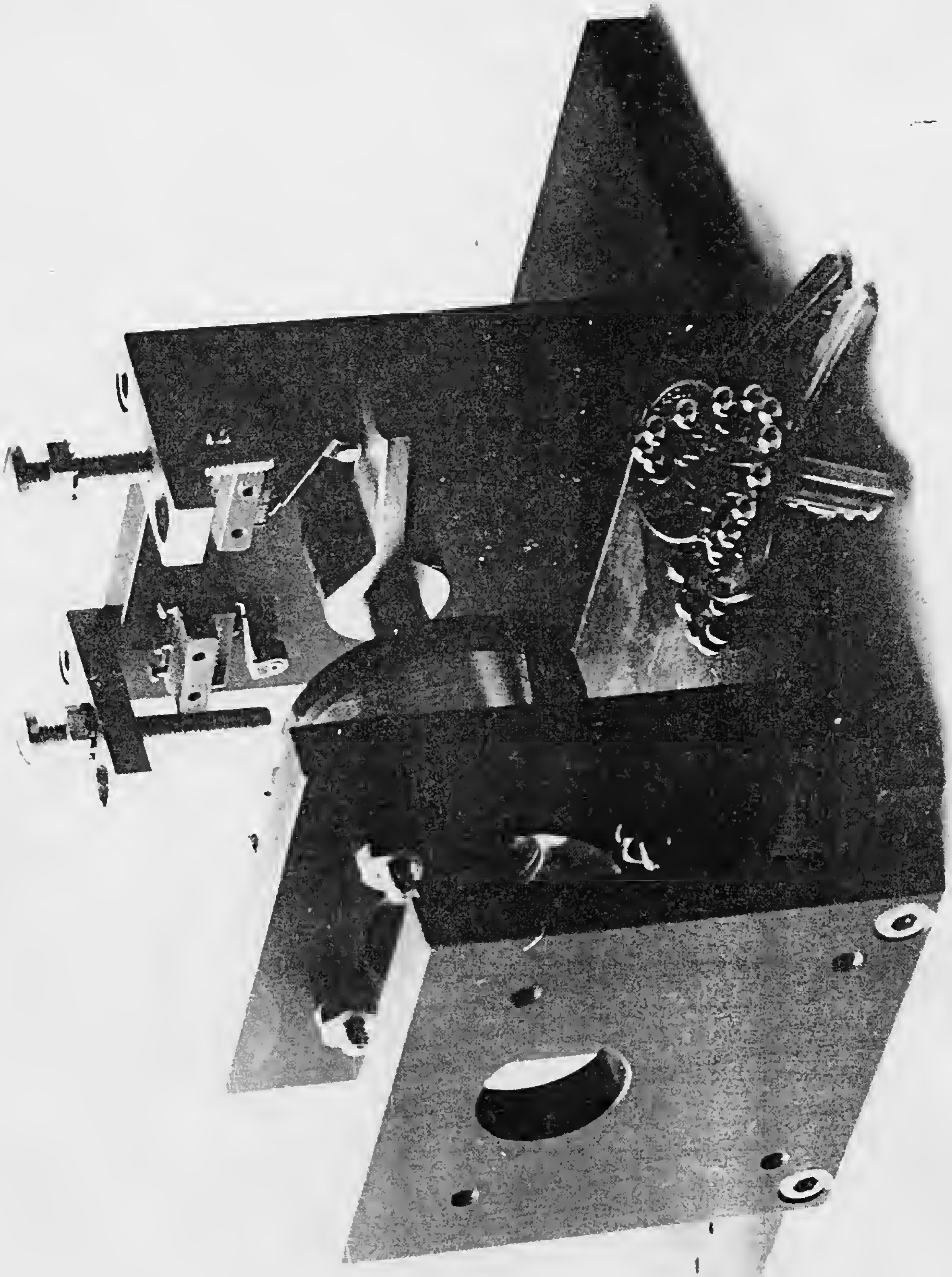


EXHIBIT A-1 Limit Switch

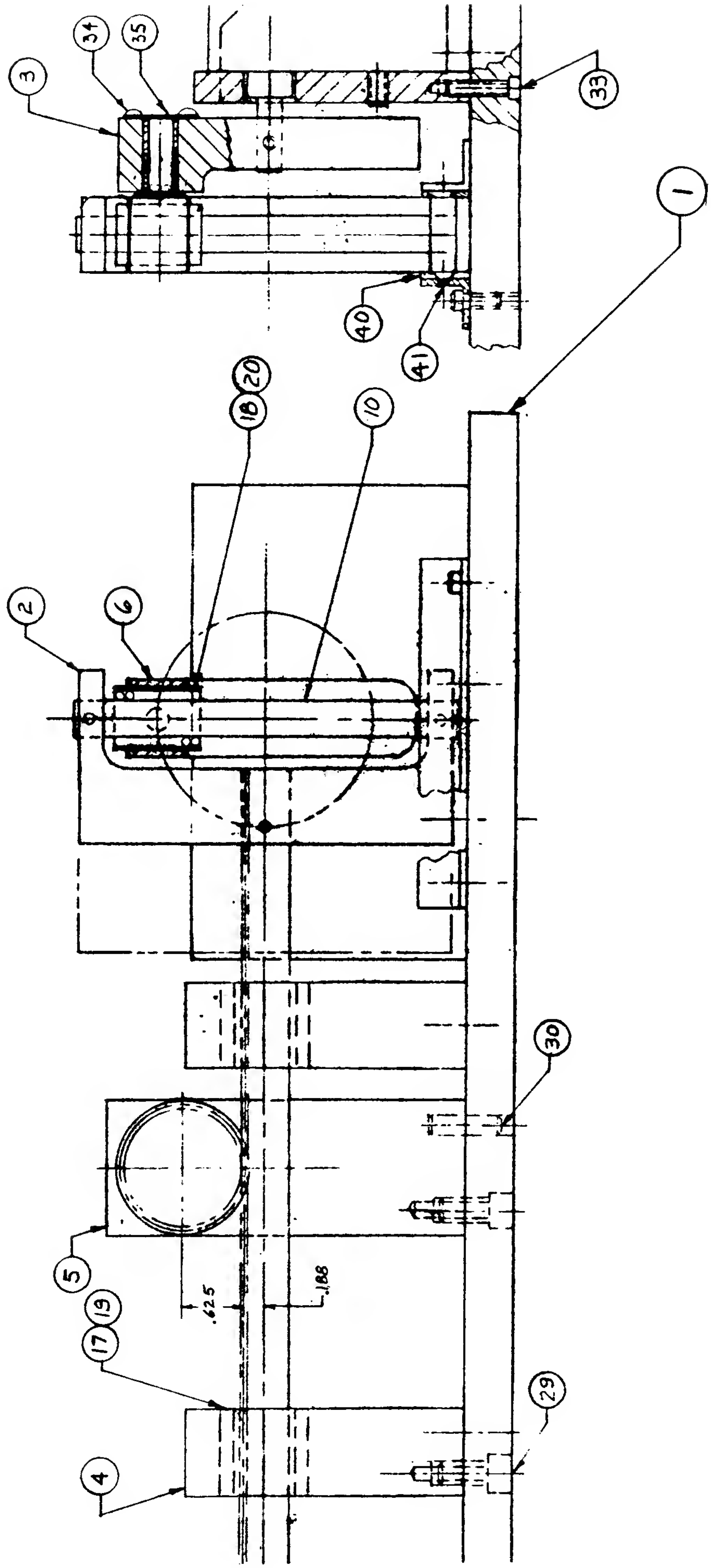


EXHIBIT A-2a Detail Drawing, First Scotch Yoke

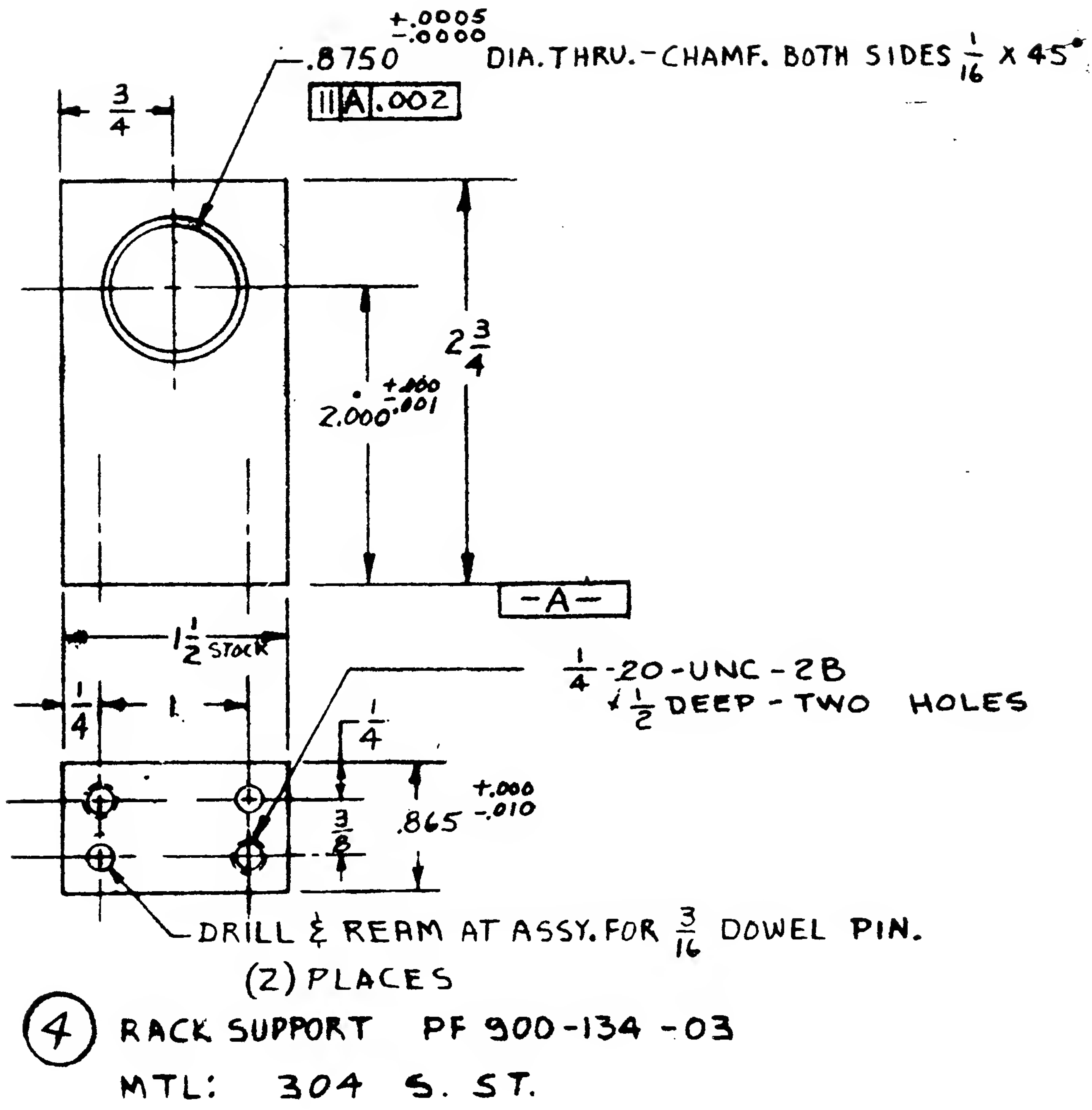


EXHIBIT A-2b Detail Drawing, First Scotch Yoke

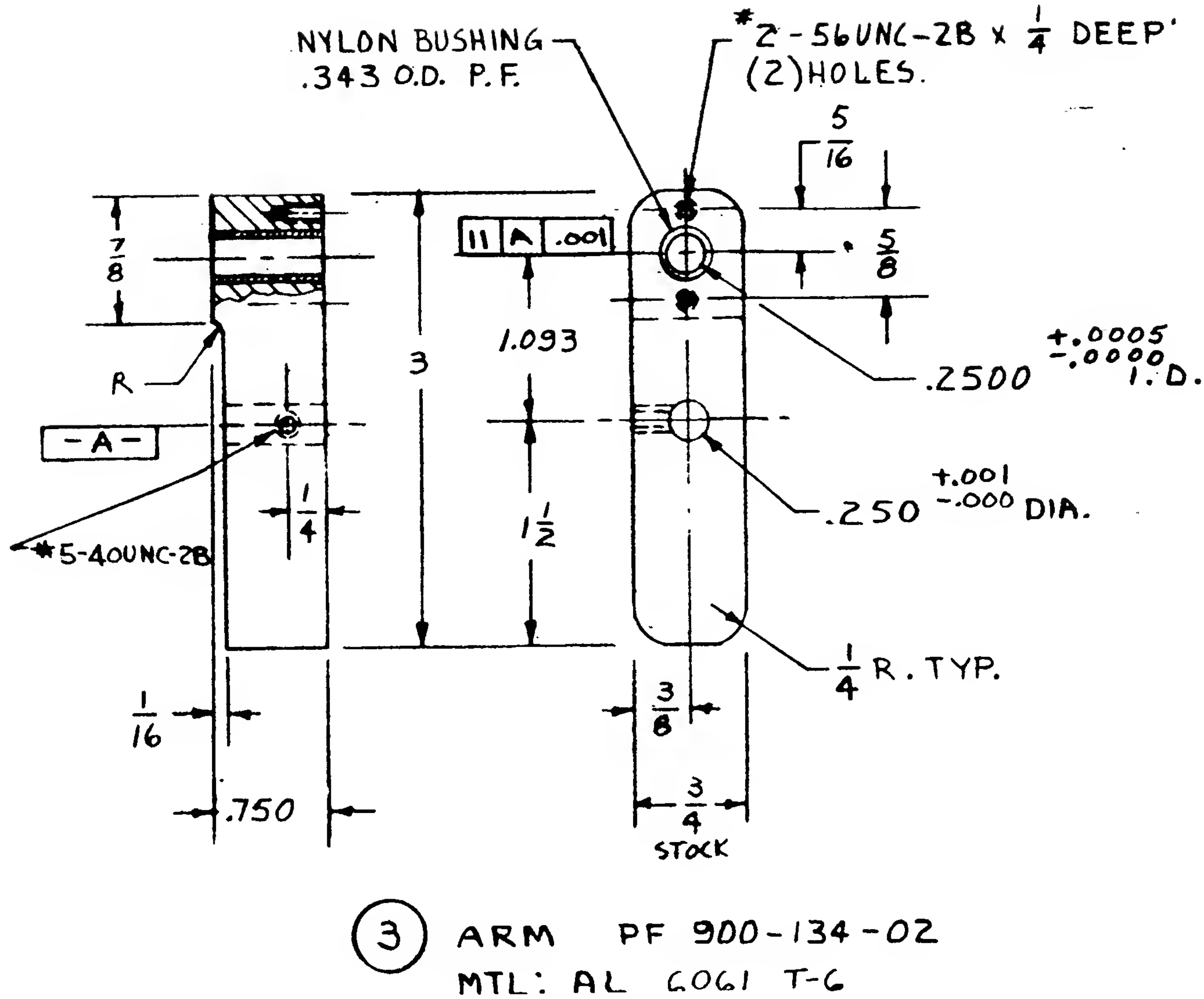


EXHIBIT A-2c Detail Drawing, First Scotch Yoke

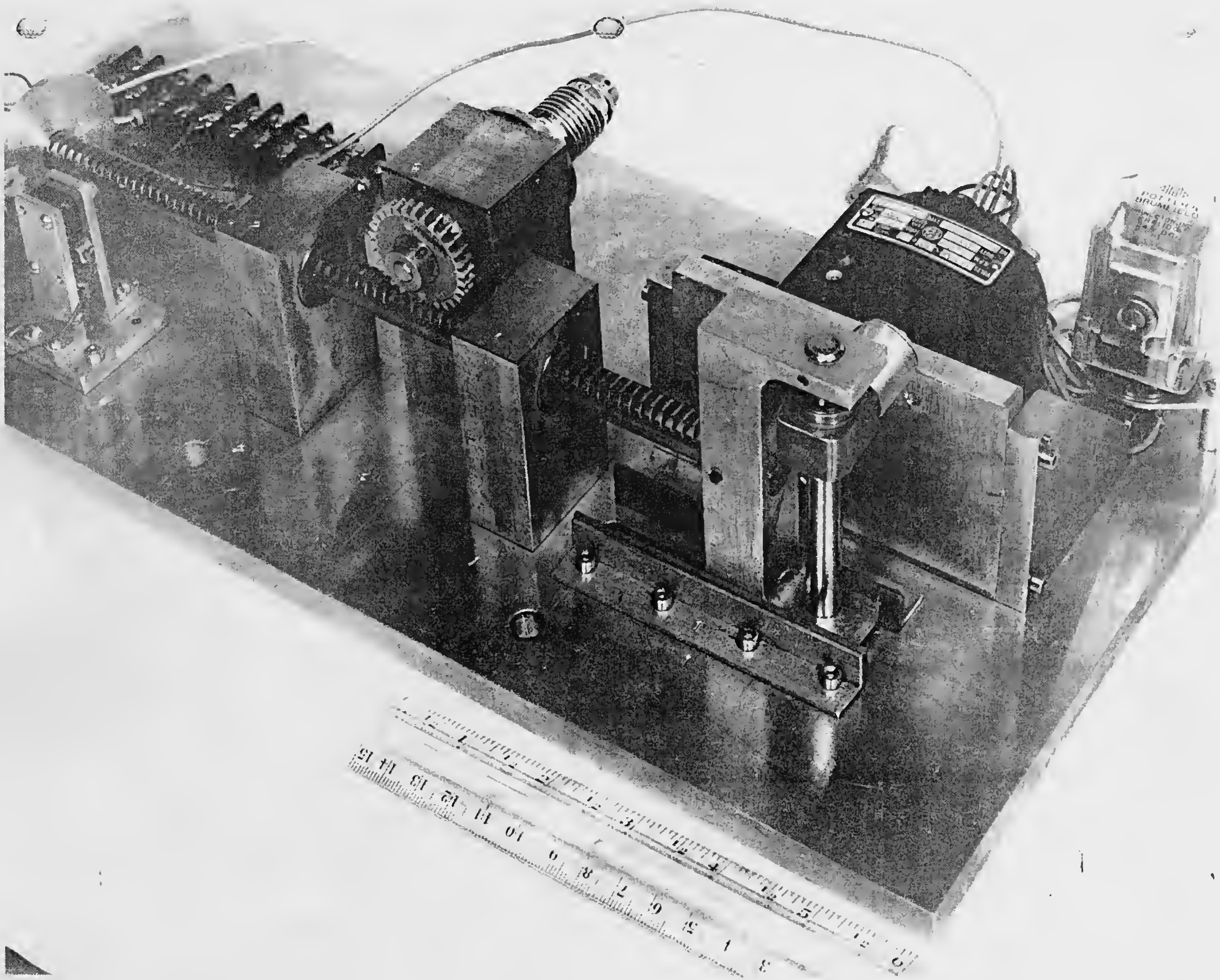


EXHIBIT A-3 First Scotch Yoke

The Type K-2 motor is characterized by laminations extending through to periphery of motor. This construction is especially well suited for motors which must be totally enclosed because it provides the most effective cooling with the greatest amount of magnetic iron.

BODINE fractional/horsepower

stock
MOTORS
Immediate Delivery

MODEL TYPE

Model Type	Reducer Output: Speed Rpm	Reducer Output: Torque In.-Oz.	Reducer Output: Gear Ratio	Type Frame	Ball Bearing Motors: Model No.	Ball Bearing Motors: Catalog No.
*CAPACITOR (3-LEAD)—Normal Slip—115 V.—60 CY.— Totally Enclosed	0.9	120	1800:1	KCI-22RM	842	B8192E-1800M
	1.9	100	900:1		841	B8192E-900M
	2.8	95	600:1		840	B8192E-600M
	5.7	95	300:1		839	B8192E-300M
	9.3	93	180:1	KCI-23RM	838	B8194E-180M
	14.0	62	120:1		837	B8194E-120M
	23.0	37	72:1		836	B8194E-72M
	56.0	18	30:1		835	B8194E-30M
	93	16.0	18:1	KCI-26RM	834	B8200E-18M
	140	10.0	12:1		833	B8200E-12M
	280	5.7	6:1		832	B8200E-06M
*CAPACITOR (3-LEAD)—High Slip—115 V.—60 CY.— Totally Enclosed	0.7	120	1800:1	KCI-22RM	863	B8262E-1800M
	1.4	100	900:1		862	B8262E-900M
	2.1	95	600:1		861	B8262E-600M
	4.1	81	300:1		860	B8262E-300M
	7	65	180:1	KCI-23RM	859	B8264E-180M
	10	43	120:1		858	B8264E-120M
	40	12	30:1		857	B8264E-30M
	67	14.0	18:1	KCI-26RM	856	B8270E-18M
	200	5.2	6:1		855	B8270E-06M
*SYNCHRONOUS CAPACITOR (3-LEAD)—115 V.—60 CY.— Totally Enclosed	1	120	1800:1	KYC-22RM	820	B8122E-1800M
	2	100	900:1		819	B8122E-900M
	3	95	600:1		818	B8122E-600M
	6	95	300:1	KYC-26RM	817	B8130E-300M
	10	65	180:1		816	B8130E-180M
	15	43	120:1		815	B8130E-120M
	25	26	72:1		814	B8130E-72M

MODEL TYPE

Model Type	Reducer Output: Speed Rpm	Reducer Output: Torque In.-Oz.	Reducer Output: Gear Ratio	Type Frame	Ball Bearing Motors: Model No.	Ball Bearing Motors: Catalog No.
*CAPACITOR (3-LEAD)—Normal Slip—115 V.—60 CY.— Totally Enclosed	0.9	110	1800:1	KCI-22RC	831	B8192E-1800C
	1.4	105	1200:1		830	B8192E-1200C
$34.21	1.9	95	900:1		829	B8192E-900C
	5.7	75	300:1		828	B8192E-300C
	9.3	70	180:1		827	B8192E-180C
	14.0	46	120:1		826	B8192E-120C
	23.0	26	72:1		825	B8192E-72C
	56	14.0	30:1	KCI-23RB	824	B8194E-30B
	93	8.6	18:1		823	B8194E-18B
	140	5.7	12:1		822	B8194E-12B
	280	2.8	6:1		821	B8194E-06B
*CAPACITOR (3-LEAD)—High Slip—115 V.—60 CY.— Totally Enclosed	0.7	110	1800:1	KCI-22RC	854	B8262E-1800C
	1.1	105	1200:1		853	B8262E-1200C
	1.4	95	900:1		852	B8262E-900C
	2.1	85	600:1		851	B8262E-600C
	4.1	75	300:1		850	B8262E-300C
	7.0	49	180:1		849	B8262E-180C
	10.0	32	120:1		848	B8262E-120C
	17.0	18	72:1		847	B8262E-72C
	40	10	30:1	KCI-23RB	846	B8264E-30B
	67	6	18:1		845	B8264E-18B
	100	4	12:1		844	B8264E-12B
	200	2	6:1		843	B8264E-06B
*SYNCHRONOUS CAPACITOR (3-LEAD)—115 V.—60 CY.— Totally Enclosed	1.0	110	1800:1	KYC-22RC	813	B8122E-1800C
	1.5	105	1200:1		812	B8122E-1200C
	2.0	95	900:1		811	B8122E-900C
	3.0	85	600:1		810	B8122E-600C
	6.0	54	300:1		809	B8122E-300C
	10.0	32	180:1		808	B8122E-180C
	15.0	21	120:1		807	B8122E-120C
	25.0	12	72:1		806	B8122E-72C
	60	6.6	30:1	KYC-23RB	805	B8124E-30B
	100	4.0	18:1		804	B8124E-18B
	150	2.6	12:1		803	B8124E-12B
	200	2.0	9:1		802	B8124E-09B
	300	1.3	6:1		801	B8124E-06B

table capacitor and container of lubricant supplied with each motor.

EXHIBIT A-4 Drive Motor (Bodine Catalogue Page)

Sheet 1 of 2

type K-2 motors

The Type K-2 frame delivers considerably more power than motors of clock type construction, and is especially adapted for use on instruments, control devices, and other equipment requiring relatively low output, and steady, trouble-free service.

TYPE K-2

- Instantly Reversible
- Synchronous or Non-synchronous
- Can be Stalled Indefinitely

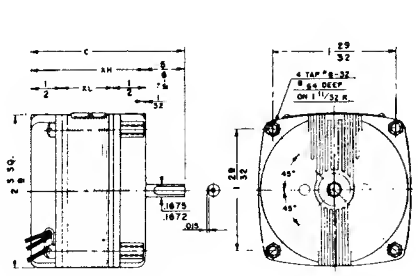
Type K-2 motors are totally enclosed, having only a small bushed opening through which the terminal leads are brought out. End shields are die cast from zinc base alloy and are carefully fitted to stator rings by machined rabbeted joints. Oil lubricated ball bearings, standard for all rotor shafts, are carefully selected and electronically inspected for defects, quietness and cleanliness. Laminations are assembled between die cast clamping rings, and two rivets at each corner hold the stator assembly firmly together.

MODEL TYPE

Speed Rpm	Torque In.-Oz.	Type Frame	Model No.	Catalog No.
*CAPACITOR (3-LEAD)—Normal Slip 115 V.—60 CY.—Totally Enclosed				
1550	1.4	KCI-23	705	B8194E
1550	2.4	KCI-26	706	B8200E
*CAPACITOR (3-LEAD)—High Slip 115 V.—60 CY.—Totally Enclosed				
1200	1.1	KCI-23	707	B8264E
1200	1.8	KCI-26	708	B8270E
*SYNCHRONOUS CAPACITOR (3-LEAD) 115 V.—60 CY.—Totally Enclosed				
1800	.35	KYC-23	701	B8124E
1800	.6	KYC-26	703	B8130E
3600	.18	KYC-23	702	B8138E
3600	.3	KYC-26	704	B8144E

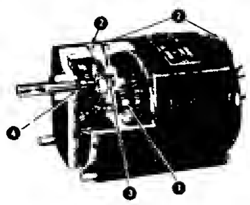
* Suitable capacitor and container of lubricant supplied with each motor.

• DIMENSIONS



Frame	C	XH	XL	Shipping Wt. Oz.
K-23	2 7/16	1 13/16	13/16	32
K-26	3 3/32	2 15/32	1 15/32	44

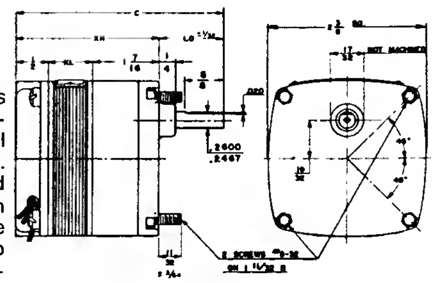
TYPE K-2 with Helical Gear Reduction



1. Ball Bearing
2. Oil Holes
3. Ball Thrust Bearing
4. Flange Type Thrust Bearing

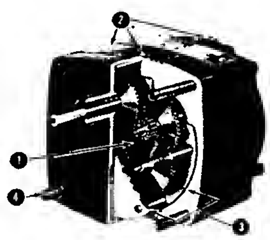
Designed for continuous duty applications requiring sturdy construction and exceptional reliability. All gearing is of the helical type, accurately hobbed to A. G. M. A. standards. Helical gearing increases load carrying capacity, reduces noise through smoother tooth action and produces more uniform pitch line velocity, as compared to spur-gears. Gear train is made up of individual gear and steel pinion assemblies, running and supported on hardened, accurately ground steel studs pressed into the gear housing to exact center distances. A laminated bakelite gear meshes with rotor pinion in the first stage. Gears in the intermediate positions are either bronze or steel. Grease lubricated gearing.

• DIMENSIONS



Frame	C	XH	XL	Shipping Wt. Oz.
K-22RM	3 5/8	2 5/8	11/16	42
K-23RM	3 3/4	2 3/4	13/16	42
K-26RM	4 13/32	3 13/32	1 15/32	54

TYPE K-2 with Spur Gear Reduction

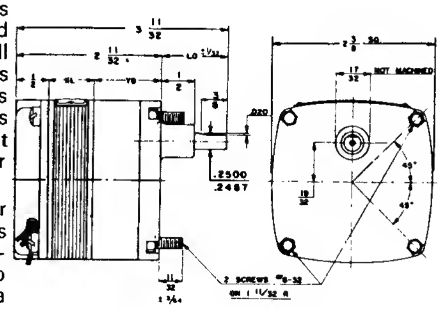


1. Steel Rotor Shaft Pinion
2. Oil Holes
3. Gear Reduction Assembly
4. Mounting Screw

This Type K-2 speed reducer motor employs a small, but rugged, built-in spur gear speed reducer. The motor shaft rotates in ball bearings, while the slow speed drive shaft is supported in a sleeve bearing. All pinions are steel; gears traveling at the higher speeds are made of bakelite for quietness; those at lower speeds of bronze to carry greater torques.

The gear and drive shaft bearings in the gear housing are lubricated by grease which has been selected for operation within the ordinary temperature range (about 40° F.). Two of the four case holding screws are extra long so that they may be passed into or through the mounting surface to hold the motor in place.

• DIMENSIONS



Frame	XL	YB-	Shipping Wt. Oz.
K-22RC	11/16	1 5/32	38
K-23RB	13/16	1 1/32	38

● Dimensions are for reference only and are correct at date of publication. For latest data request certified dimension sheet.

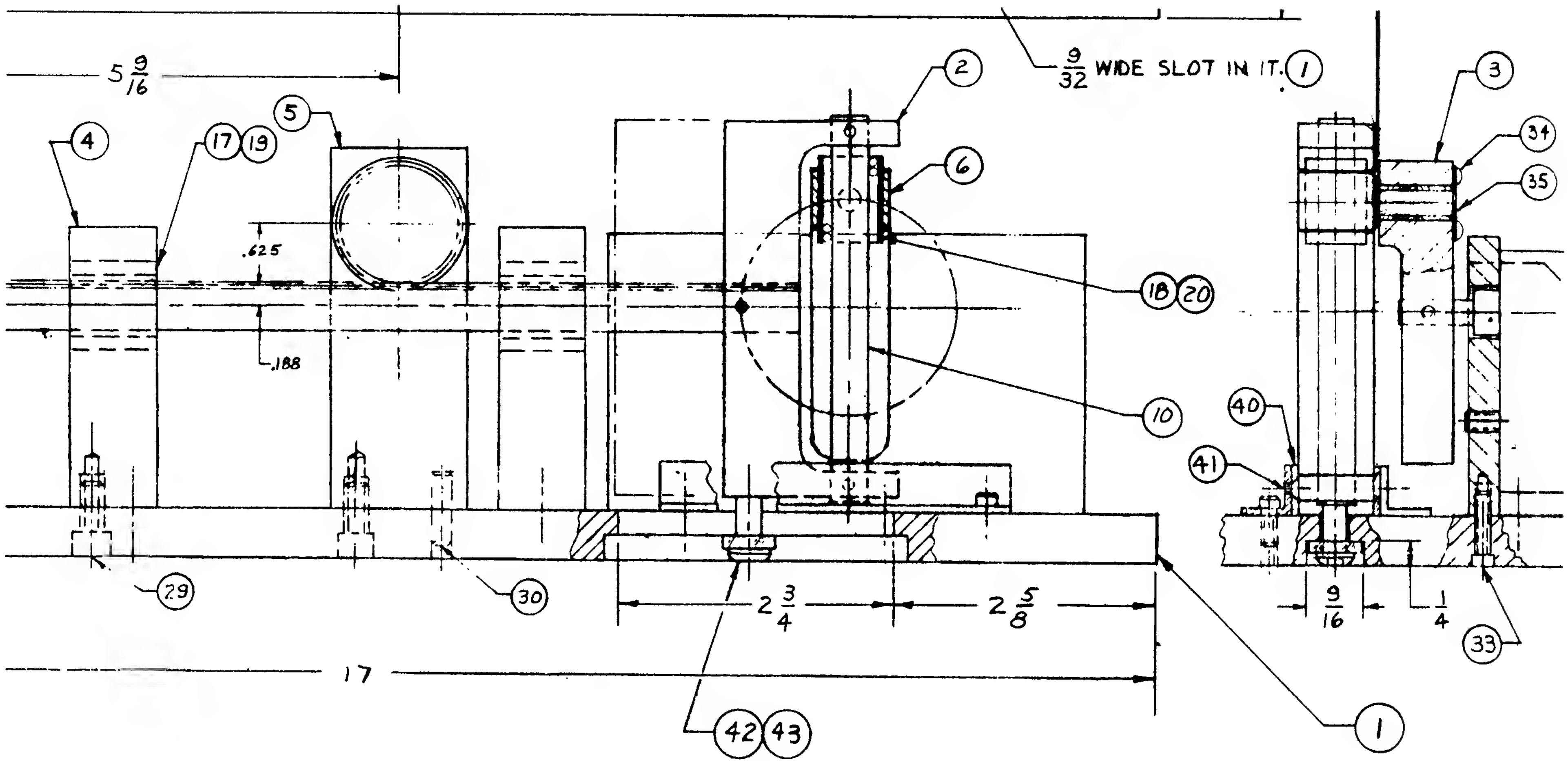


EXHIBIT A-5 Drawing of Modified First Scotch Yoke
Note the addition of items 42 and 43 and the slot to accommodate them.

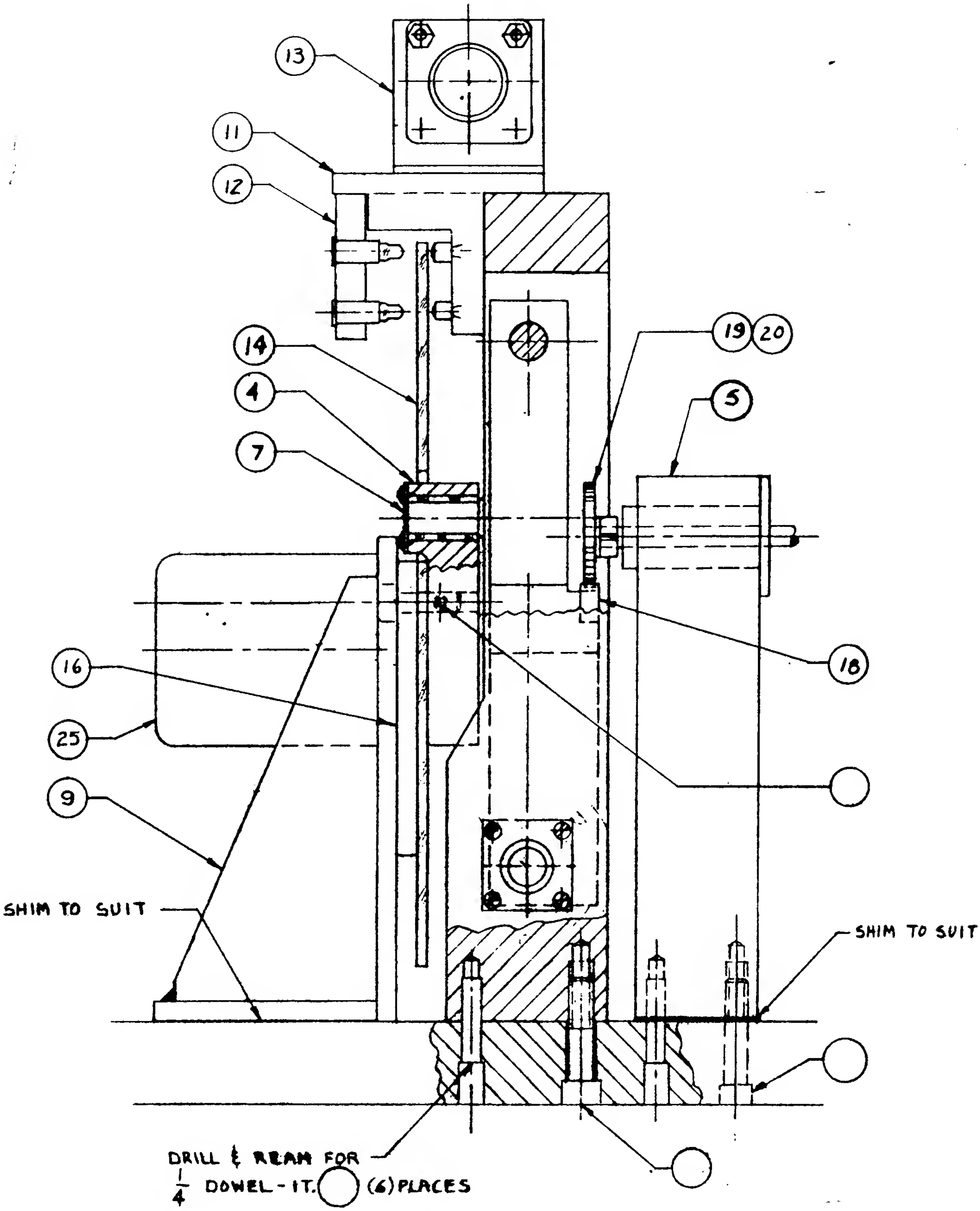


EXHIBIT A-6a Detail Drawing, Second Scotch Yoke

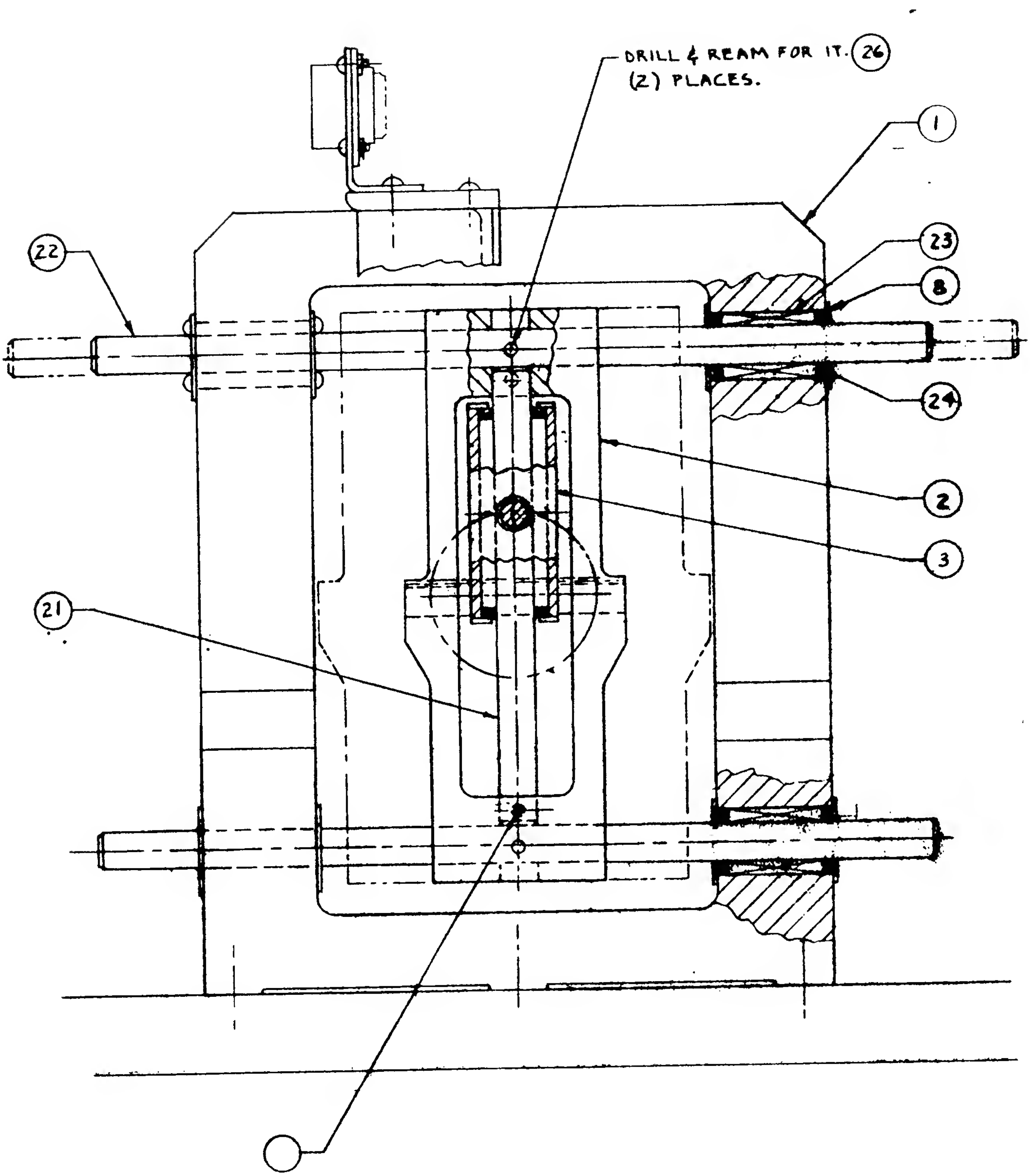


EXHIBIT A-6b Detail Drawing, Second Scotch Yoke

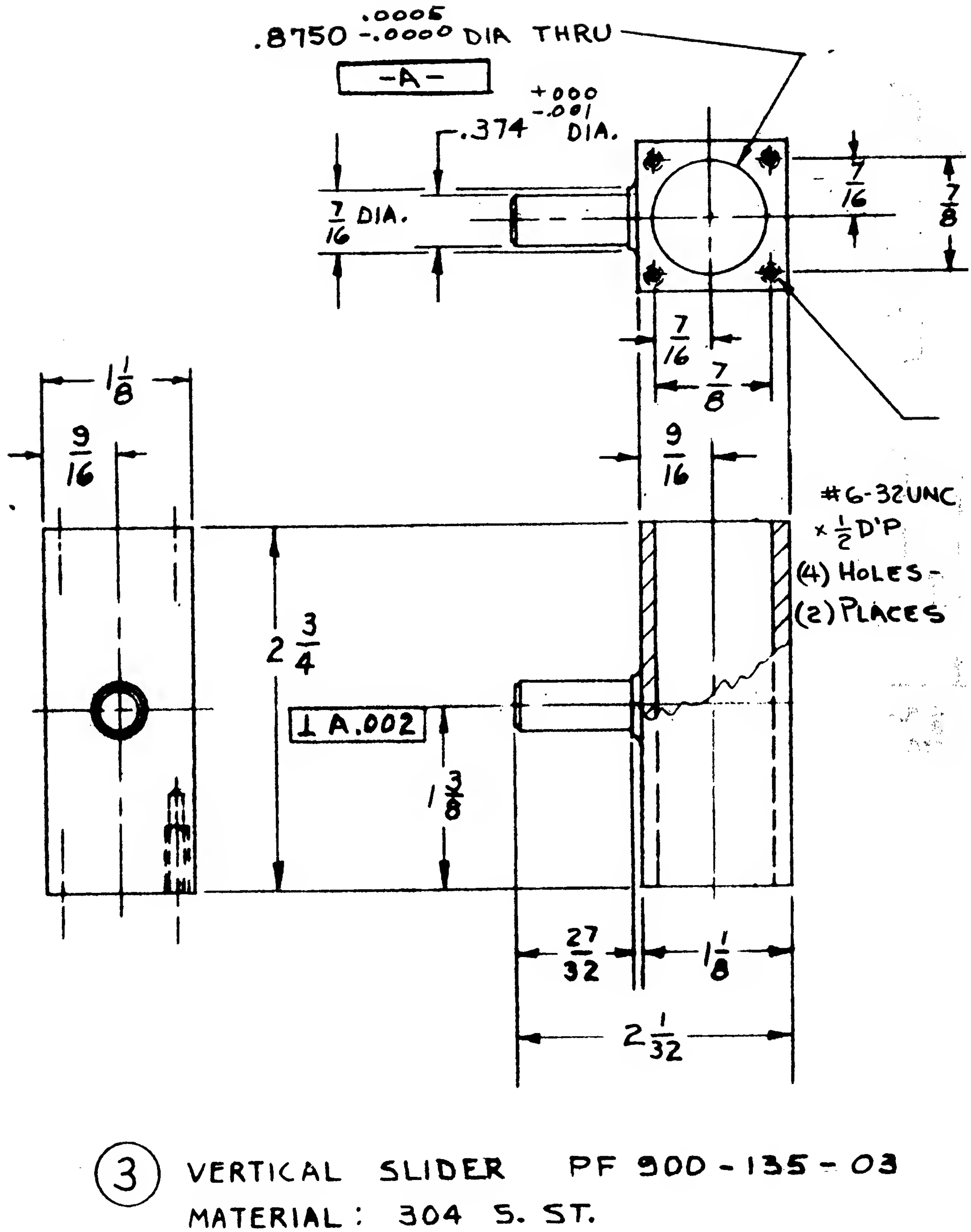
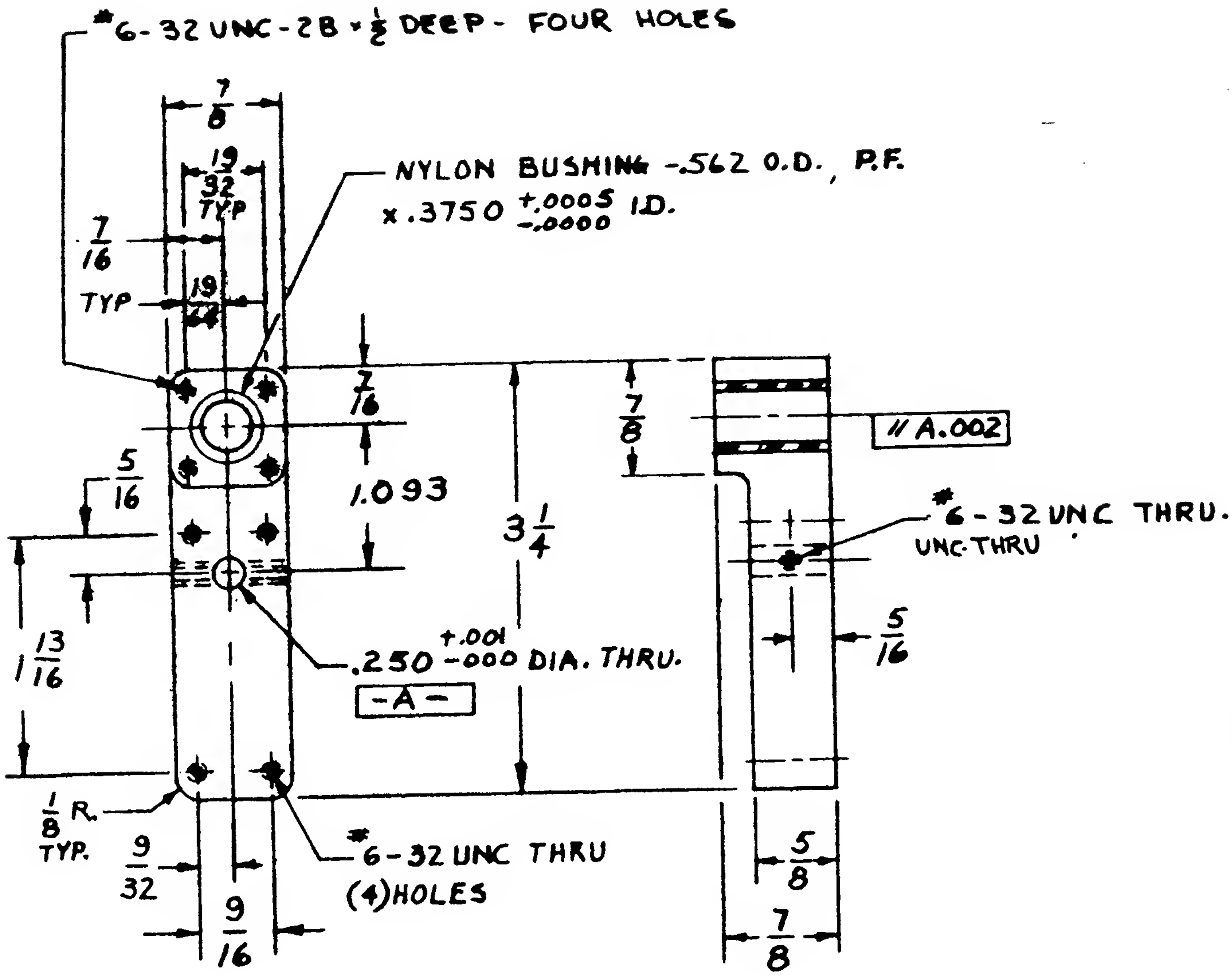


EXHIBIT A-6c Detail Drawing, Second Scotch Yoke



4 ARM PF 900-135-04
MATL: 6061-T6 ALUM.

EXHIBIT A-6d Detail Drawing, Second Scotch Yoke

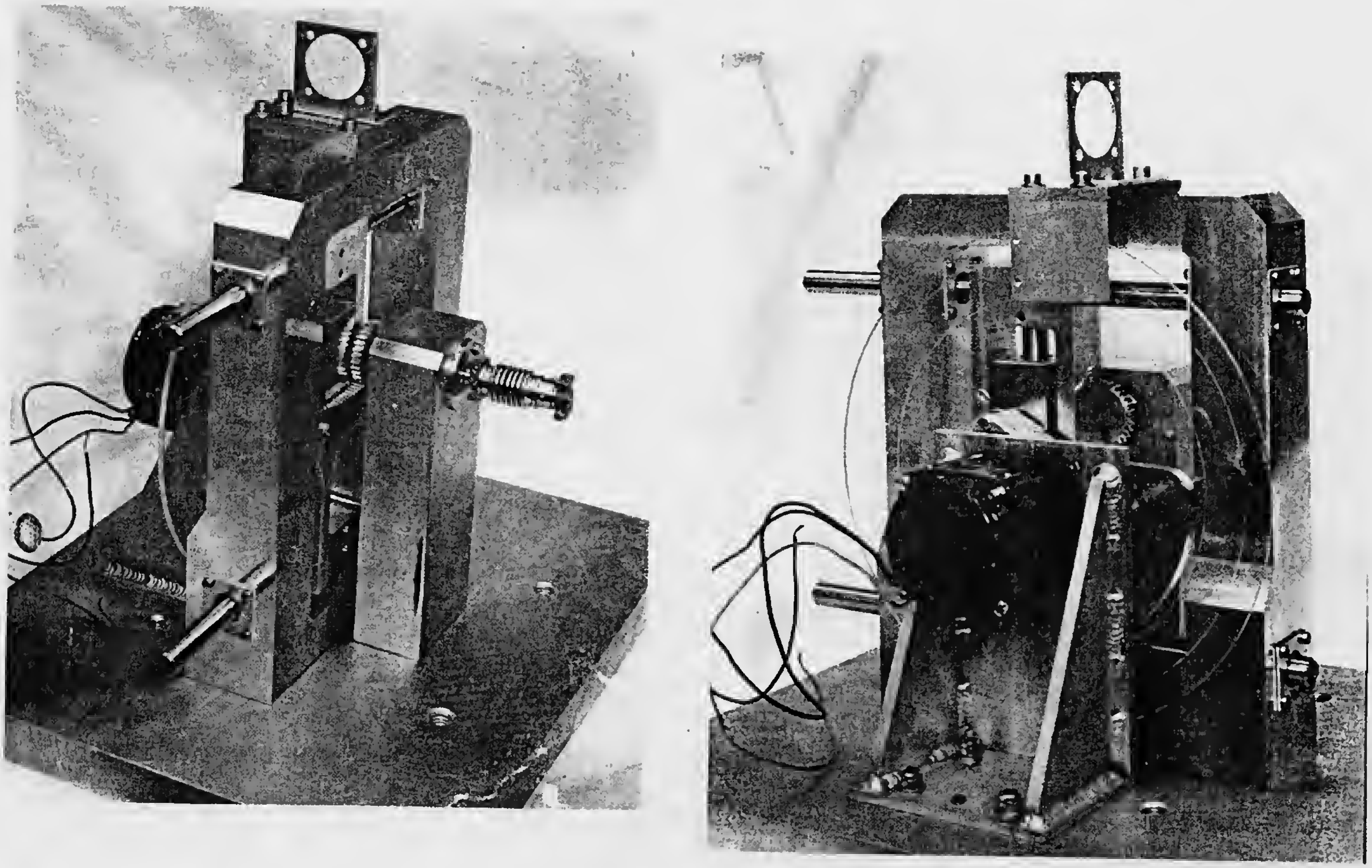
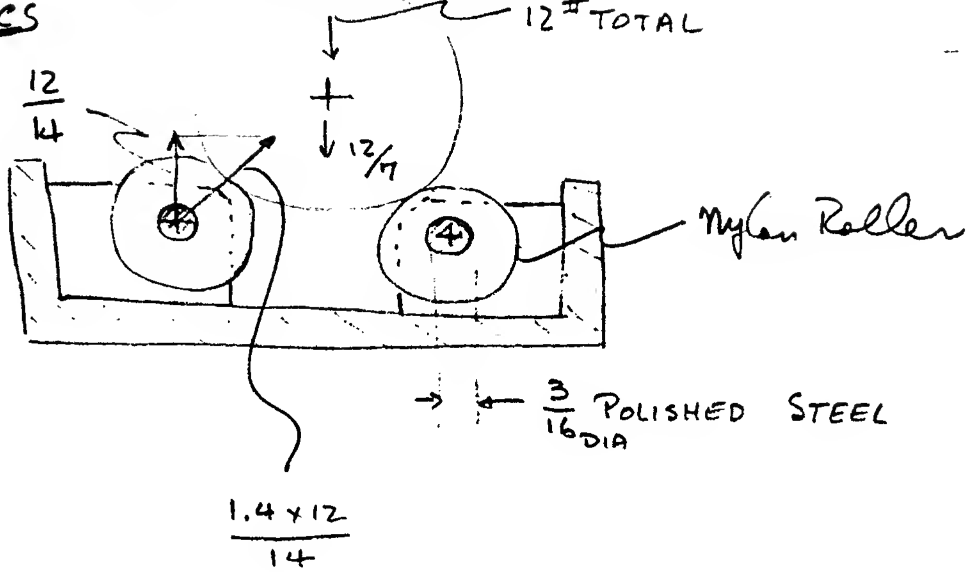


EXHIBIT A-7 Photographs, Second Scotch Yoke

FLIP COIL DRIVE

TORQUE CALCS



Assume μ = .4

Friction Force = $.4 \times \frac{1.4 \times 12}{14}$

FRICTION TORQUE/WHEEL = $.4 \times \frac{1.4 \times 12}{14} \times .093$

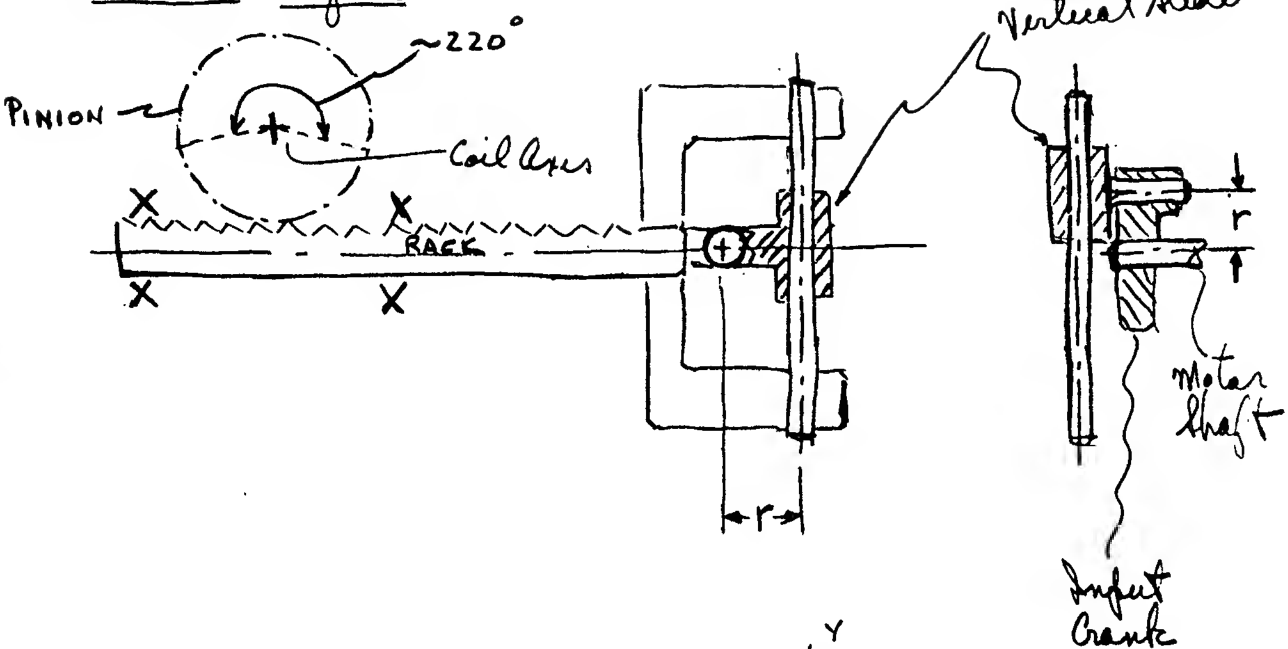
TOTAL FRICTION TORQUE = $\frac{.4 \times 1.4 \times 12 \times .093 \times \cancel{14}}{\cancel{14}}$ #"

= .626 #" or 10.02 oz in

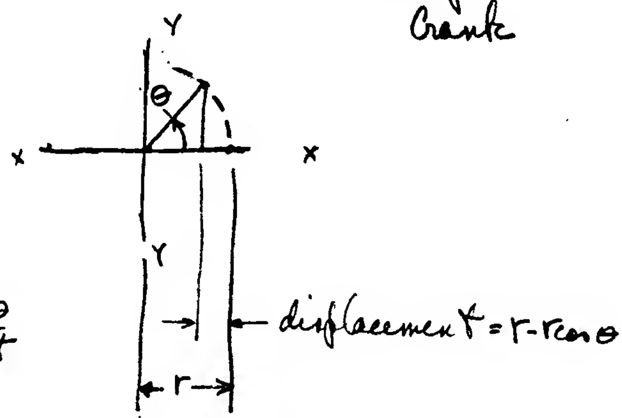
FLIP- COIL - DRIVE

TORQUE CALCS

Inertial Torque



Initial Position of Input Crank :



1- displacement = $s = r - r\cos\theta$

2- velocity = $v = \frac{ds}{dt} = -r(-\sin\theta)\frac{d\theta}{dt}$

$= r\omega \sin\theta$

3- acceleration = $a = \frac{dv}{dt} = r\{\omega^2\cos\theta + \alpha\sin\theta\}$

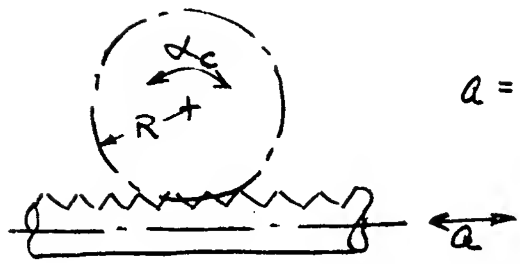
i.e. when $\theta = 0$, accel is "normal" --- $r\omega^2$

$\theta = 90°$, accel is tangential --- $r\alpha$

for constant ω, accel = $r\omega^2\cos\theta$

FLIP COIL DRIVE

For Coil



$$a = R\alpha_c$$

$$\alpha_c = \frac{r\omega^2\cos\theta}{R}$$

$$\alpha_c = \frac{d\omega_c}{dt} = \frac{r}{R}\omega^2\cos\theta$$

$$d\omega_c = \frac{r}{R}\omega^2\cos\theta\, dt = \frac{r}{R}\omega\cos(\omega t)\,\omega dt$$

Integrating
$$\int d\omega_c = \frac{r}{R}\omega\int_0^t \cos(\omega t)\cdot\omega dt$$

$$\omega_c = \frac{r}{R}\omega\left[\sin\omega t\right]_0^t = \frac{r}{R}\omega\sin\omega t$$

$$\omega_c = \frac{d\theta_c}{dt} = \frac{r}{R}\omega\sin\omega t$$

$$\int d\theta_c = \frac{r}{R}\int_0^t \sin\omega t\cdot\omega dt$$

$$\theta_c = -\frac{r}{R}\left[\cos\omega t\right]_0^t = -\frac{r}{R}\left[\cos\omega t - 1\right]$$

$$= \frac{r}{R}\left[1-\cos\omega t\right]$$

Motor Rotates at 3 rpm
$\therefore\ \omega = 2\pi/20\,\text{sec} = \pi/10$ rads/sec

at $t = 0$ $\quad\theta = 0$
at $t = 5$ secs $\quad\theta = \frac{r}{R}\left[1-\cos\frac{\pi}{2}\right] = \frac{r}{R}$
at $t = 10$ secs $\quad\theta = \frac{r}{R}\left[1-\cos\pi\right] = 2\frac{r}{R}$

checks

FLIP COIL DRIVE

Inertial Torque Reaction of Coil

$$T = J_{MASS}\,\alpha_c = \frac{1}{2}\,\frac{W_{COIL}}{g}\,r_{COIL}^2 \times \frac{r}{R}\,\omega^2\cos\theta$$

$W_{COIL} = 12^{\#} \times 16 = 192$ oz

$r_{coil} = .687''$

$g = 32 \times 12 = 384$ "/sec²

$R = .625$ } Assume 24P; $1\frac{1}{4}$ P.D. PINION

$r = 1.093''$

Max torque occurs at $\theta = 0$ and $\theta = \pi$

$$T = \frac{1}{2} \times \frac{192}{384} \times (.687)^2 \times \frac{1.093}{.625} \times \frac{\pi^2}{100}$$

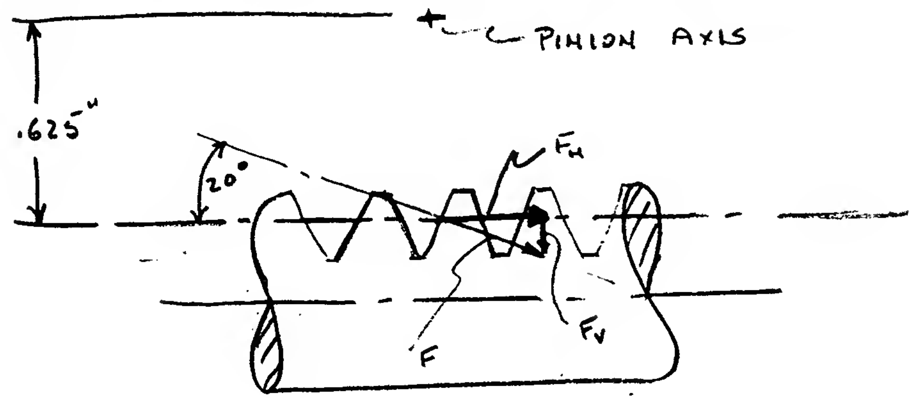
$= .5 \times .5 \times .472 \times 1.748 \times .098$

$= .0428$ oz inches max

This is approximately .4 % of the friction load and may be neglected in any consideration of power requirement or gear tooth and bearing load.

FLIP COIL

Gear Tooth Load & BEARING LOAD CALCS



$F_H = \dfrac{10.02 \text{ oz in}}{.625 \text{ in}}$

$F_H = 16.03$ oz

$= 1^{\#}$ say

and $F_V = \dfrac{10.02}{.625} \tan 20°$

$F_V = \dfrac{10.02 \times .364}{.625} = 5.83$ oz.

$= .36^{\#}$

Select Thomson Ball Bushings to support rack

Design life = 10,000 hrs ⇒ 600,000 minutes

Travel life = $(2r) \times 2 \times$ R.P.M. $\times$ Design life

$= 4 \times 1.093 \times 3 \times 6 \times 10^5 = 78.696 \times 10^5$

Load Correction factor K_L (see Thomson Catalog) = .65

For a soft stainless steel rack (303) the load correction factor K_H = .03 (see Thomson Catalog)

FLIP COIL

weight of yoke & rack

Load Capacity required per bushing is: $\frac{F_V}{2} + \frac{\text{weight}}{2} = \frac{.36}{2} + \frac{1.0}{2} = .68$

$$\text{Rolling Load Rating} = \frac{\text{Load Capacity Req'd}}{K_L \times K_H} = \frac{.68}{.65 \times .03} = 34.87^{\#}$$

THOMSON BALL BUSHING A-81420 ($\frac{1}{2}$" BORE) is O.K.

We can use PIC Round Rack AG-17 / 24 PITCH

(see page 332 PIC CAT.)

Yoke

Bushing feels $F_H = 1^{\#}$ as a constant load

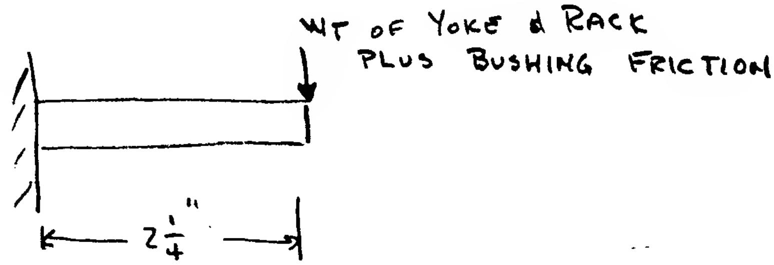
$$\text{Rolling Load Rating} = \frac{\text{Load Cap.}}{K_L \times K_H} = \frac{1}{.65 \times .9} = 1.7^{\#}$$

use Thomson hardened shaft Rc 58/63

THOMSON BALL BUSHING A-61014

Deflection of Rack

(assume 3/8" dia)



$$\delta = \frac{W l^3}{3EI} = \frac{1 \times 11.4}{3 \times 30 \times 10^6 \times .049 \times .02} \approx .0001$$

δ = negligible

FLIP COIL

Coil Angle of Twist

$$\phi = \frac{T L}{J G}$$

T = applied torque = .626 "#

L = length : assume full length of 144"

$$J = \frac{\pi D^4}{32} = 0.098 \times \left(1\tfrac{3}{8}\right)^4 = 3.572 \text{ "}^4$$

G = 178,000 #/□"

$$\phi = \frac{.626\text{"\#} \times 144\text{"}}{3.572^{\text{"}4} \times 178,000 \text{ \#/□"}}$$

ϕ = .00014 radians

Twist caused by friction.

Note: Torque used here was friction torque. Inertial torque reaction was small enough to be neglected.

To cover an expanding line of BALL BUSHINGS, *revised Data Sheets will be issued periodically giving the latest data on the sizes currently in production. If your latest Data Sheet is over 4 or 5 months old, we suggest you contact your local representative (shown on back cover of catalog) or write the factory requesting the latest information.*

Attach Middle Page to Inside Back Cover of New Catalog No. 4

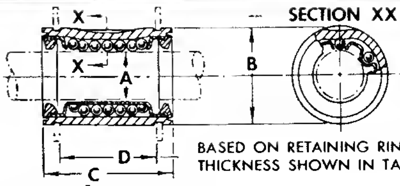
SUPPLEMENTARY DATA SHEET No. 7

Engineering and Price Data Effective January 1, 1963

SUPERSEDES Supplementary Data Sheet No. 6a

BALL BUSHING ENGINEERING SPECIFICATIONS

APPLIES TO TABLES 2, 3 and 4



BASED ON RETAINING RING THICKNESS SHOWN IN TABLE 18

The following data cover the Precision Series A, the Super-Precision Series XA, the Commercial Grade Series B, the Adjustable Diameter Series ADJ, and the Open Type Series OPN BALL BUSHINGS. The five types are made to the same basic dimensions with varying degrees of precision. They all have the same free-rolling characteristics. For more information refer to Index on inside back cover of the catalog.

Table #2 — PRECISION SERIES A — Dimensions & Load Ratings

TYPE	BALL BUSHING NUMBER	WORKING BORE A		OUTSIDE DIAMETER B		LENGTH C		DISTANCE BETWEEN RETAINING RINGS D		MAXIMUM PERMISSIBLE SHAFT DIA. ***	RECOMMENDED HOUSING BORE NORMAL FIT		RECOMMENDED HOUSING BORE PRESS FIT		BALL DIAMETER	NUMBER OF BALL CIRCUITS	BUSHING WEIGHT	RATED LOAD* STATIC	RATED LOAD* ROLLING **	BALL BUSHING NUMBER
		INCHES	TOL.	INCHES	TOL.	INCHES	TOL.	INCHES	TOL.	INCHES	INCHES	TOL.	INCHES	TOL.	INCHES		POUNDS	POUNDS	POUNDS	
SERIES A	A-4812	.2500		.5000		.750		.437		.2490	.5000		.4990		1/16	3	.02	22	13	A-4812
	A-61014	.3750		.6250		.875		.562		.3740	.6250		.6240		1/16	4	.06	38	21	A-61014
	A-81420	.5000	+ .0000 — .0005	.8750	+ .0000 — .0004	1.250	+ .000 — .015	.875	± .010	.4990	.8750	+ .0005 — .0000	.8740	+ .0005 — .0000	3/32	4	.08	72	46	A-81420
	A-122026	.7500		1.2500		1.625		1.062		.7490	1.2500		1.2490		1/8	5	.21	162	109	A-122026
	A-162536	1.000		1.5625		2.250		1.625		.9990	1.5625		1.5615		5/32	5	.38	262	202	A-162536
	A-203242	1.2500	+ .0000 — .0006	2.0000	+ .0000 — .0005	2.625		1.875		1.2490	2.0000		1.9983		3/16	6	1.1	465	344	A-203242
	A-243848	1.500	+ .0000 — .0006	2.375	+ .0000 .0005	3.000	+ .000 — .020	2.250		1.4989	2.375		2.3733	+ .001 — .000	7/32	6	1.43	695	535	A-243848
	A-324864	2.000	+ .0000 — .0008	3.000	+ .0000 — .0006	4.000		3.000	± .015	1.9987	3.000	+ .001 — .000	2.9982		1/4	6	2.75	1100	850	A-324864
	A-406080	2.500	+ .0000 — .0010	3.750	+ .0000 — .0008	5.000	+ .000 — .025	3.750		2.4985	3.750		NOT NORMALLY RECOMMENDED		5/16	6	5.5	1710	1380	A-406080
	A-487296	3.000	+ .0000 — .0012	4.500	+ .0000 — .0010	6.000	+ .000 — .030	4.500		2.9983	4.500				3/8	6	9.5	2460	2000	A-487296
	A-6496128	4.000	+ .000 — .002	6.000	+ .0000 — .0012	8.000	+ .000 — .040	6.000	± .020	3.9976	6.000				1/2	6	20.2	4400	3800	A-6496128

*Based on a shaft hardness of Rockwell 60C. **Based on a travel life of 2 million inches. (See Catalog Page 18.) ***For normal fit slightly larger shafts may be used with caution. (See Catalog Page 16.)

STAINLESS STEEL: Series A and XA BALL BUSHINGS can be obtained made entirely of stainless steel. They are identified by the suffix SS following the part number (Example–XA-81420-SS). They are available only in the smaller sizes up to and including A- and XA-162536-SS.

Table #3 — SUPER PRECISION SERIES XA — Dimensions & Load Ratings

TYPE	BALL BUSHING NUMBER	WORKING BORE A		CONCENTRICITY	OUTSIDE DIAMETER B		LENGTH C		DISTANCE BETWEEN RETAINING RINGS D		MAXIMUM PERMISSIBLE SHAFT DIAMETER	RECOMMENDED HOUSING BORE NORMAL FIT†		RECOMMENDED HOUSING BORE PRESS FIT	BALL DIAMETER	NUMBER OF BALL CIRCUITS	BUSHING WEIGHT	RATED LOAD* STATIC	RATED LOAD* ROLLING **	BALL BUSHING NUMBER
		INCHES	TOL.	T.I.R.	INCHES	TOL.	INCHES	TOL.	INCHES	TOL.	INCHES	INCHES	TOL.	INCHES	INCHES		POUNDS	POUNDS	POUNDS	
SERIES XA	XA-4812	.2500			.5000		.750		.437		.2495	.5000		NOT NORMALLY RECOMMENDED	1/16	3	.02	22	13	XA-4812
	XA-61014	.3750			.6250		.875		.562		.3745	.6250			1/16	4	.06	38	21	XA-61014
	XA-81420	.5000	+ .0000 — .0003	.0005	.8750	+ .0000 — .0004	1.250	+ .000 — .015	.875	± .010	.4995	.8750	+ .0005 — .0000		3/32	4	.08	72	46	XA-81420
	XA-122026	.7500			1.2500		1.625		1.062		.7495	1.2500			1/8	5	.21	162	109	XA-122026
	XA-162536	1.000			1.5625		2.250		1.625		.9995	1.5625			5/32	5	.38	262	202	XA-162536
	XA-203242	1.2500			2.0000	+ .0000 — .0005	2.625		1.875		1.2495	2.0000			3/16	6	1.1	465	344	XA-203242
	XA-243848	1.500	+ .0000 — .0004	.0010	2.375	+ .0000 — .0005	3.000	+ .000 — .020	2.250		1.4994	2.375			7/32	6	1.43	695	535	XA-243848
	XA-324864	2.000			3.000	+ .0000 — .0006	4.000		3.000	± .015	1.9994	3.000	+ .001 — .000		1/4	6	2.75	1100	850	XA-324864
	XA-406080	2.500	+ .0000 — .0005		3.750	+ .0000 — .0008	5.000	+ .000 — .025	3.750		2.4993	3.750			5/16	6	5.5	1710	1380	XA-406080
	XA-487296	3.000	+ .0000 — .0006	.0015	4.500	+ .0000 — .0010	6.000	+ .000 — .030	4.500		2.9992	4.500			3/8	6	9.5	2460	2000	XA-487296
	XA-6496128	4.000	+ .0000 — .0010	.0020	6.0000	+ .0000 — .0012	8.000	+ .000 — .040	6.000	± .020	3.9988	6.000			1/2	6	20.2	4400	3800	XA-6496128

*Based on a shaft hardness of Rockwell 60C. **Based on a travel life of 2 million inches. (See Catalog Page 18.) †For extreme precision, tolerance may be reduced.

Table #4 — COMMERCIAL GRADE SERIES B — Dimensions & Load Ratings

In lots of 250 or more

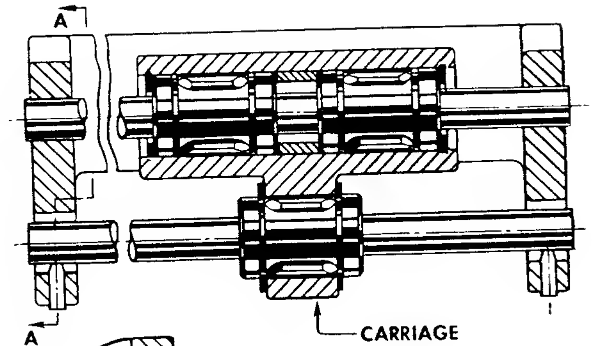
TYPE	BALL BUSHING NUMBER	WORKING BORE A		OUTSIDE DIAMETER* B		LENGTH C		DISTANCE BETWEEN RETAINING RINGS D		MAXIMUM PERMISSIBLE SHAFT DIAMETER NORMAL FIT	MAXIMUM PERMISSIBLE SHAFT DIAMETER PRESS FIT	RECOMMENDED HOUSING BORE NORMAL FIT		RECOMMENDED HOUSING BORE PRESS FIT		BALL DIAMETER	NUMBER OF BALL CIRCUITS	BUSHING WEIGHT	RATED LOAD** STATIC	RATED LOAD** ROLLING ***	BALL BUSHING NUMBER
		INCHES	TOL.	INCHES	TOL.	INCHES	TOL.	INCHES	TOL.	INCHES	INCHES	INCHES	TOL.	INCHES	TOL.	INCHES		POUNDS	POUNDS	POUNDS	
SERIES B	B 4812	.250		.500		.750		.437		2495	.2495	.5000		.4980		1/16	3	.02	19	11	B-4812
	B-61014	.3750		.6250	+ .0000 — .0015	.875	+ .000 — .020	.562	± .010	.3745	.3740	.6250		.6230	+ .0005 — .0000	1/16	4	.06	33	18	B-61014
	B-81420	.500		.875		1.250		875		.4995	.4990	.8750		.8730		3/32	4	.08	61	39	B 81420
	B 122026	.750	+ .002 — .000	1.250		1.625	+ .000 — .025	1.062		.7495	.7490	1 2500	+ .001 — .000	1.2475		1/8	5	.21	138	93	B-122026
	B 162536	1.000		1.5625		2 250		1.625		.9995	.9990	1.5625		1.5600		5/32	5	37	222	172	B 162536
	B 203242	1 250		2.0000	+ .000 — .002	2.625	+ .000 — .030	1.875	± .015	1.2495	1.2490	2.0000		1.9970	+ .001 — .000	3/16	6	1 1	400	292	B 203242
	B-243848	1.500		2.375		3.000		2.250		1.4994	1.4989	2.375		2.372		7/32	6	1.4	590	455	B-243848

*Slight out-of-roundness may result from the heat-treatment of Series B bearings, making it difficult to measure the true O.D. The bearing will return substantially to its original roundness when inserted into the recommended housing bore for either normal or press fit. **Based on a shaft hardness of Rockwell 60C. ***Based on a travel life of 2 million inches (see page 18).

Mounting Arrangements

IF PARALLEL SHAFTS ARE USED IN PRECISION APPLICATIONS, ACCURATE ALIGNMENT IS IMPORTANT.

When BALL BUSHINGS are used to mount a carriage for linear travel, it is frequently necessary to prevent the carriage from rotating. Two parallel shafts may be used for this purpose. In precision applications where little or no play can be tolerated and the shafts are fitted closely to the bearings, care must be exercised to assure parallelism of the shafts or roughness and possible damage to the bearings may result. Shaft parallelism can be obtained by accurate location



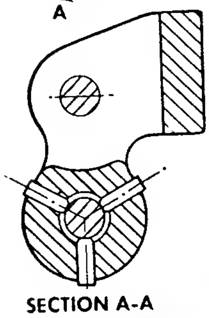


FIG. 16—PARALLEL SHAFTS WITH ADJUSTMENT

A linear travel carriage mounted on three BALL BUSHINGS, two of which ride on a fixed shaft and the third on an adjustable parallel shaft which is retained at each end by three set screws in an oversize hole. The adjustments can be used to take out play due to diametral clearance between the shaft and bearings as well as for shaft alignment.

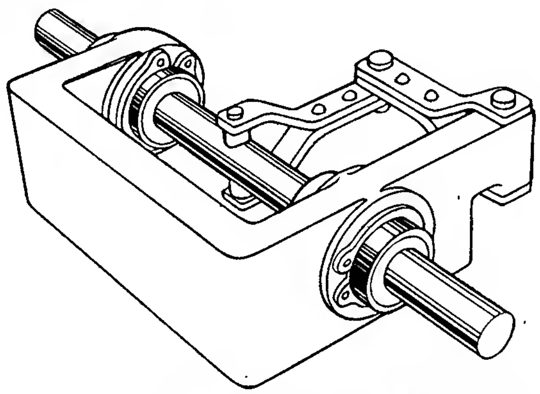


FIG. 17—SINGLE SHAFT WITH LINKAGE GUIDE

A shaft mounted on two BALL BUSHINGS with an intermediary linkage to prevent relative rotation between the shaft and the bearing housing during linear travel.

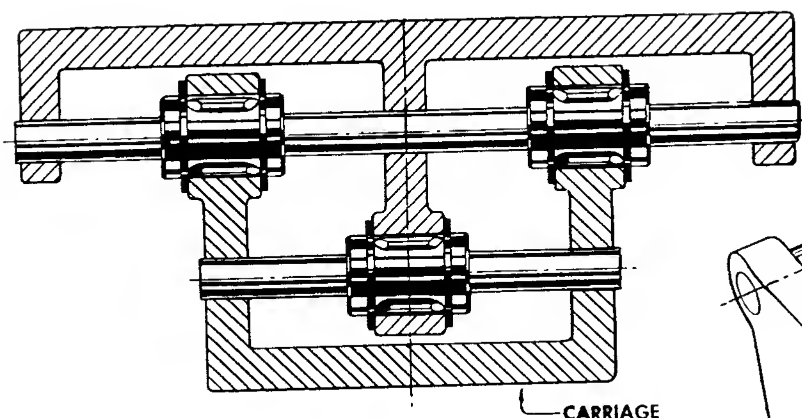
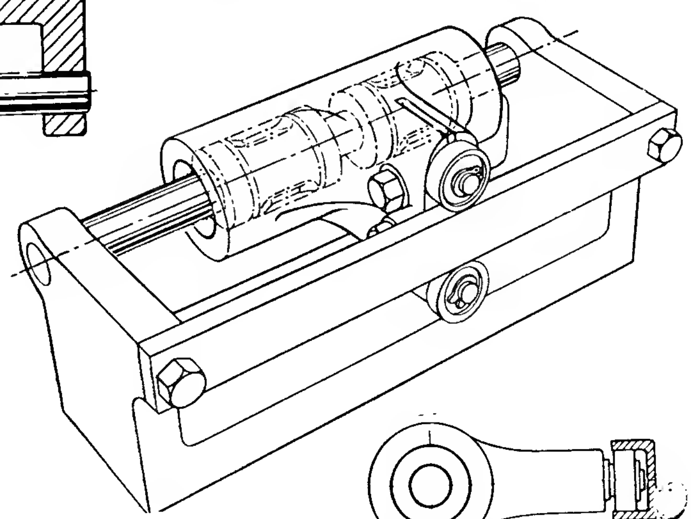


FIG. 18—LONG AND SHORT PARALLEL SHAFTS

A linear travel carriage mount, featuring two BALL BUSHINGS mounted on the carriage and riding on a long fixed shaft with an intermediate support, and a third BALL BUSHING mounted on the fixed structure and riding on a parallel shaft which is mounted on the carriage.

FIG. 19—CARRIAGE ON SINGLE SHAFT IN COMBINATION WITH A TORQUE ROLLER

A linear travel carriage mounted on two BALL BUSHINGS which ride on a fixed shaft, with provision for double or single rollers riding on a guide rail to prevent rotation of the carriage.



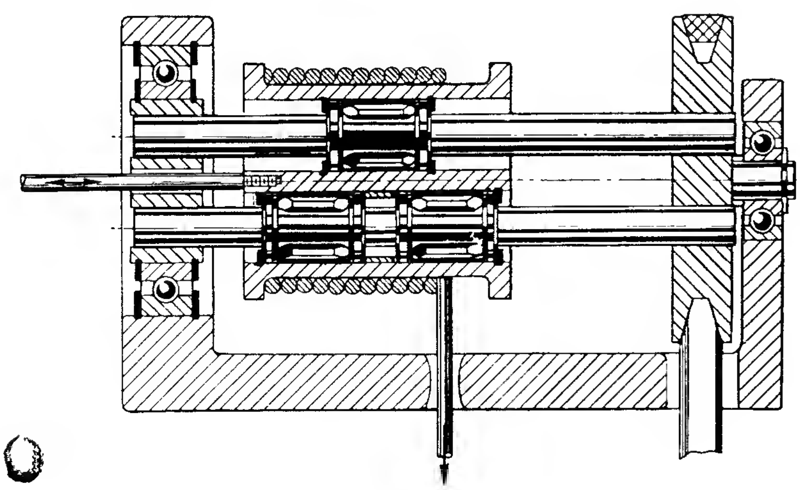
ALTERNATIVE MOUNTING USING U-CHANNEL & SINGLE ROLLER

of the shaft mounting holes or by providing an adjustment to permit proper alignment. Flexible mounting of one of the Ball Bushings or a floating arrangement of one of the shafts or Ball Bushings can sometimes be used as an alternative arrangement. *For information on flexible* Ball Bushing *mounts see Page 15.* For applications in which both rigidity and extreme precision are required, individual Ball Bushing mounting blocks are recommended. This permits precise adjustment and alignment of Ball Bushings to shafts. In many non-precision light load applications the shafts can be finished to a diameter sufficiently undersize to allow for a reasonable amount of shaft or bearing misalignment.

Instead of using two parallel shafts, it may be preferable to make use of a linkage or roller guides to prevent rotation of the carriage on a single shaft. The accompanying illustrations suggest a few of the many mounting arrangements possible with Ball Bushings.

FIG. 20—TORQUE TRANSMISSION

Torque can be transmitted to or from a free rolling linear motion by mounting the reciprocating part on a pair of parallel shafts which are secured in the rotating members.



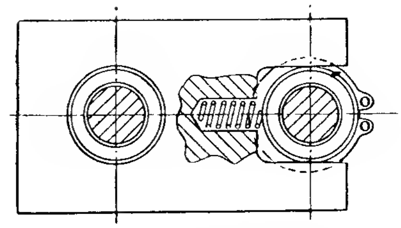


FIG. 21—SPRING LOADED BALL BUSHINGS

A floating Ball Bushing can be spring loaded in numerous ways to take out all shake or play resulting from the recommended diametral clearance between the shaft and the bearing. The spring force should be well in excess of the maximum load on the mechanism, but no more than the rolling load rating of the Ball Bushing.

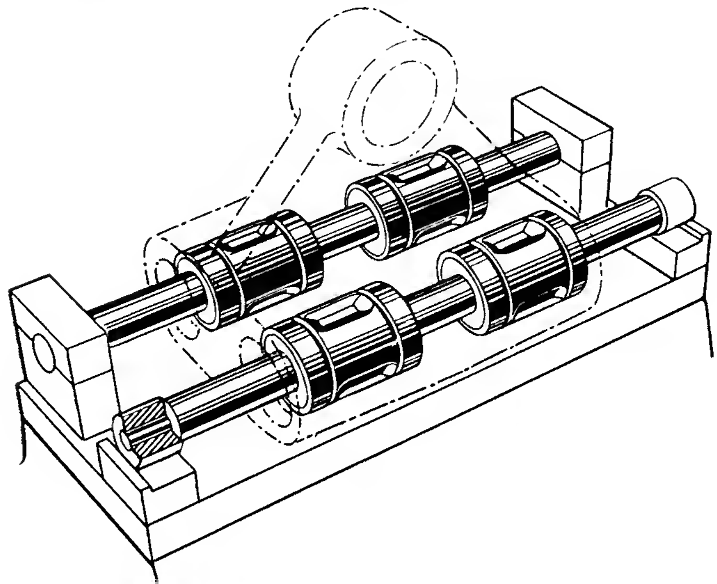


FIG. 22—FLOATING SHAFT

In applications where the load is always in one direction, one shaft can be rigidly mounted and the other allowed to float on rollers riding on hardened pads. In this arrangement, the shafts are self-aligning in one plane, but the pads must be dimensioned or shimmed to assure parallelism in the other plane.

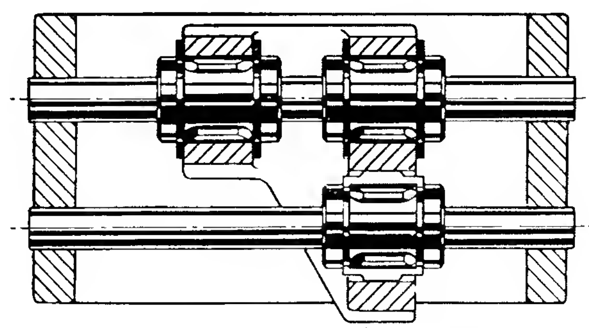


FIG. 23—PARALLEL SHAFTS WITH A RESILIENT MOUNTED BALL BUSHING

A linear travel carriage mounted on three Ball Bushings, two of which are rigidly mounted in the carriage and ride on a fixed shaft, and a third in a standard resilient mount riding on a fixed parallel shaft. See Page 15.

Load Capacity and Ball Bushing Life

THE LOAD CAPACITY OF A BALL BUSHING AND SHAFT COMBINATION IS INFLUENCED BY THE LIFE EXPECTANCY AND BY THE HARDNESS OF THE SHAFT

Life expectancy is expressed in terms of the total inches of linear movement between the BALL BUSHING and the shaft during its operating life and is known as its *Travel Life*. The shaft hardness is expressed in terms of the Rockwell "C" required for no grooving of the shaft. (see last two paragraphs under "Ball Bushing Shafts," Page 16).

The *Rolling Load Ratings* given in Tables 2 *thru* 6 on the Data Sheet are based on a shaft hardness of Rockwell 60C and a Travel Life of 2,000,000 inches. To find the Allowable Load Capacity for other conditions of Travel Life or shaft hardness, the Rolling Load Ratings must be multiplied by the appropriate load correction factors K_L and K_H obtained from Chart 1 and Chart 2 respectively. To solve for other items refer to Table 1.

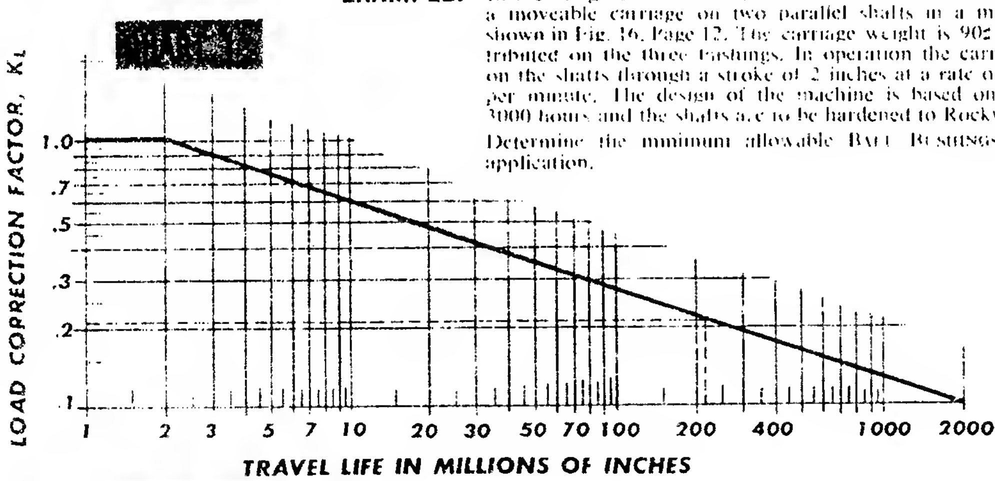
The *Static Load Ratings* given in Tables 2 *thru* 6 are based on a shaft hardness of Rockwell 60C and must be corrected by factor K_H when a softer shaft is used. They are given to indicate allowable non-Brinell loads and are to be used only in special cases where the expected Travel Life is relatively low.

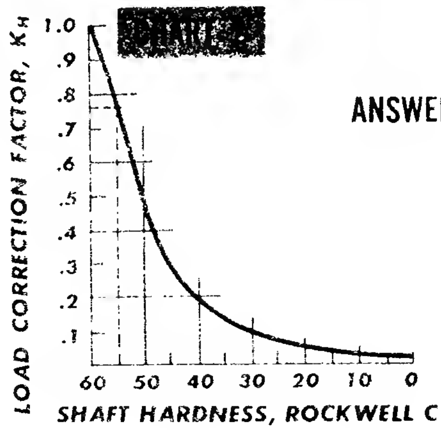
TABLE 1

If you wish to determine	And you know	Then solve for
1. Allowable load capacity	a. BALL BUSHING size b. Travel life req'd c. Shaft hardness	Allowable load capacity Rolling Load Rating x K_L x K_H
2. Travel life expectancy	a. BALL BUSHING size b. Load capacity req'd c. Shaft hardness	$K_L = \dfrac{\text{Load Capacity req'd}}{\text{Rolling Load Rating} \times K_H}$ and read travel life from Chart 1
3. Minimum allowable bushing size	a. Load capacity req'd b. Travel life req'd c. Shaft hardness	Rolling Load Capacity $= \dfrac{\text{Load Capacity Req'd}}{K_L \times K_H}$ and choose bushings with the next highest rating from Tables 2 thru 6
4. Minimum allowable shaft hardness	a. BALL BUSHING size b. Load capacity req'd c. Travel life req'd	$K_H = \dfrac{\text{Load Capacity Req'd}}{\text{Rolling Load Rating} \times K_L}$ and read shaft hardness from Chart 2

EXAMPLE: In the design of a machine, three BALL BUSHINGS are to be used to support a moveable carriage on two parallel shafts in a manner similar to that shown in Fig. 16, Page 12. The carriage weight is 90# which is equally distributed on the three bushings. In operation the carriage is to reciprocate on the shafts through a stroke of 2 inches at a rate of 300 complete cycles per minute. The design of the machine is based on an operating life of 3000 hours and the shafts are to be hardened to Rockwell 55C minimum.

Determine the minimum allowable BALL BUSHINGS size for the above application.





ANSWER: From the data given above it is known that:

$$\text{Load Capacity required per bushing} = \frac{90}{3} = 30 \text{ pounds}$$

Travel life = (2"x2) x 300 CPM x 60 min. x 3000 hrs. = 216,000,000 inches
Shaft hardness = Rockwell 55C • Referring to Table 1 it is noted that, in determining minimum allowable bushing size, the following formula is used:

$$\text{Rolling Load Rating} = \frac{\text{Load Capacity Req'd}}{K_L \times K_H} = \frac{30}{.215 \times .76} = 184 \text{ pounds}$$

Referring to the Rolling Load Ratings in Table 2 of the Data Sheet it is found that Bushing No. A-162536 for a 1" diameter shaft is the minimum size which can be used in the above application.

PLASTICS IN SLIDING BEARINGS

by **GEORGE CARLYON**
Vice President, Manufacturing
Cadillac Plastic & Chemical Co.

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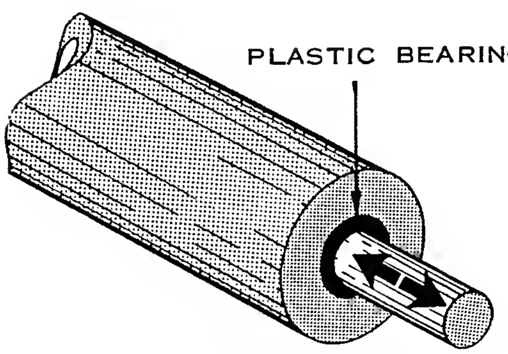
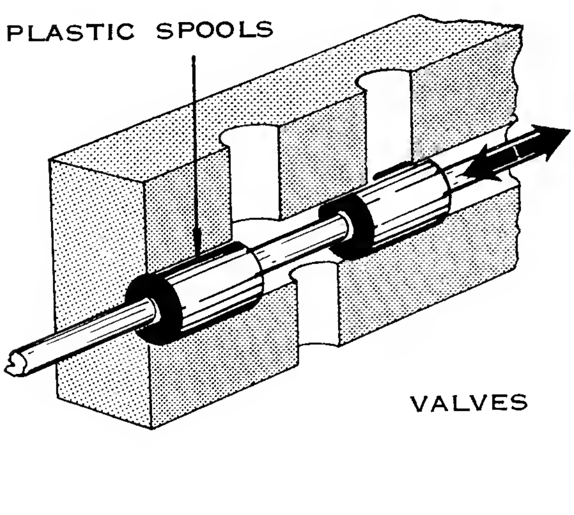
Plastics Ease Way for Sliding Bearing Design

By GEORGE CARLYON
Vice-President, Manufacturing
Cadillac Plastic & Chemical Co.
Division of Dayco Corporation



Drawings indicate range of applications in which plastics sliding bearings might prove advantageous.



THE LOW FRICTION plastics—nylon, *Teflon* TFE, and *Delrin* acetal—are well known as bearing materials in sleeve and journal applications. They may be even more advantageous in plane surface applications where relative motion is linear or reciprocal, rather than rotational. Examples of such applications include machine tool ways, elevator gibs, or slippers and actuating devices in a wide range of equipment.

The conditions under which most plane surface bearings work tend to minimize limitations of the plastic bearing materials and to favor their advantages. Speeds are, in most instances, low in comparison with those normally encountered in journal bearing design. Loading often is cyclical rather than continuous. Cooling is nearly always less of a problem than it is in journal bearing design.

Thus, the limitations of plastic bearings in comparison with metal—lower heat distortion temperature, greater creep—become less significant.

At the same time, plastics are of particular value in sliding applications because of their outstanding abrasion resistance. Plastics ways for machine tools, for example, provide long service-free life, eliminating the need to scrape and resurface ways. Ease of replacement also reduces the need to provide large or intricate wear-compensation devices.

The ability of plastics to operate without lubrication solves another of the major problems

TABLE II Design PV Limits*

Material	Velacity Ranges, fpm: 0 to 100	100 to 200	200 to 400	400 to 600
Nylon	3,000	2,600	2,200	1,800
"Delirn" acetol resin	3,000	2,600	2,200	2,100
"Teflon" TFE fluorocorbon resin (reinforced compositions)	9,600	9,600	9,600	9,600
Fobric of TFE fiber	25,000	Not recommended		

* Established for sleeve journal bearings from data compiled by E. I. Du Pont de Nemours, Inc., and Dixon Corporation. These are absolute maximums, and good practice suggests selection of values about 75% of these maximums.

in sliding bearing design. Virtually the only way to provide lubrication to linear bearing surfaces is by hydrostatic or force-fed means. Use of plastics can eliminate pumps, tubing connectors, receptacles, and maintenance attention, simplifying machine design and eliminating a number of potential trouble points. Plastics also can eliminate the mess, inconvenience, and intermittant non-lubrication problems often associated with greases.

(Initial lubrication of plastic bearings, even a wiped-off coat of oil, substantially improves operation.)

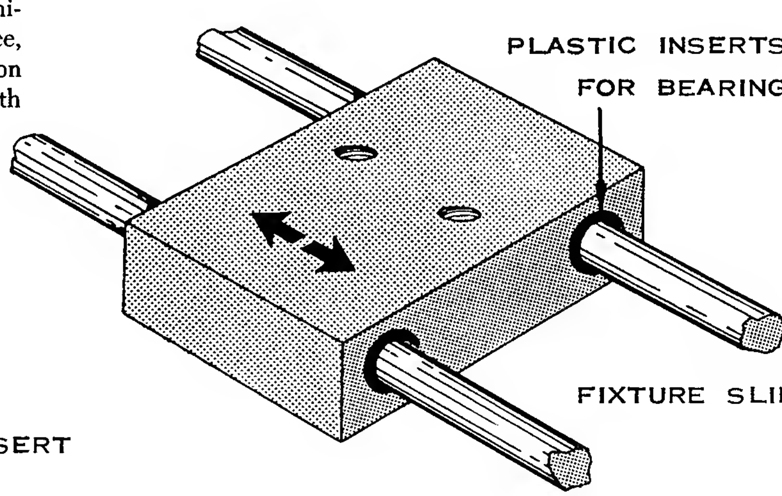
Moreover, in many such applications freedom from chattering starts is a primary consideration. Two of the plastics, TFE and acetal, offer inherent freedom from slip-stick. Metal bearing materials can offer, at best, a condition approaching no slip-stick, and then only by the use of solid film or hydrostatic lubrication.

With present information, design of all reciprocating bearings is at best a "cut and try" art. Availability of low-friction plastics in a range of standard shapes of strip, sheet, and plate makes prototype testing comparatively uncomplicated.

These plastics are of particular value to the designer in the following situations: High abrasive or corrosive ambient conditions; difficult or impossible lubrication; or where stick-slip is undesirable.

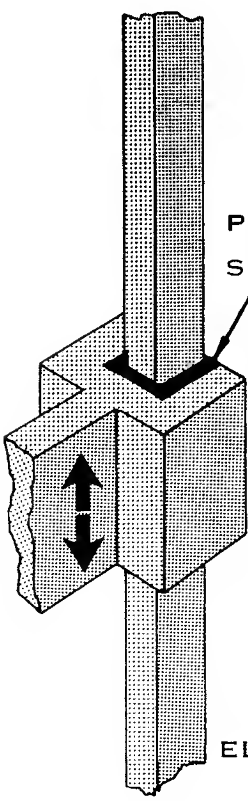
Although design criteria for plane surface bearings are comparable to those for journal bearings, the significance and

PLASTIC INSERTS FOR BEARING



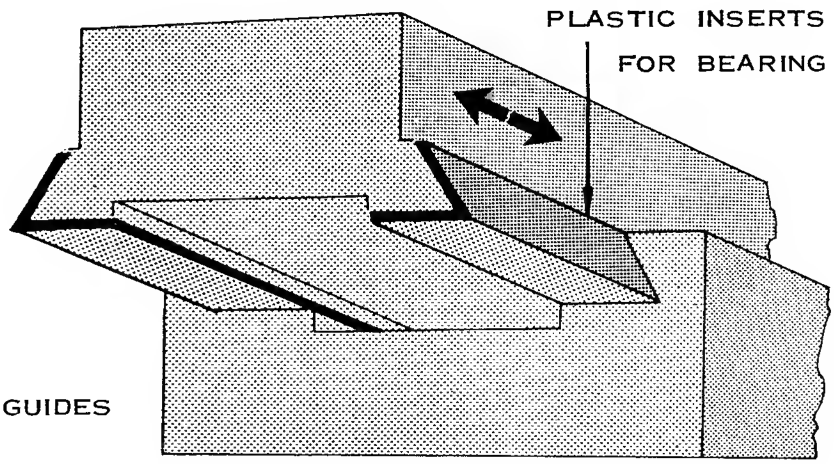
FIXTURE SLIDES

PLASTIC SLIPPER INSERT



ELEVATOR GUIDES

PLASTIC INSERTS FOR BEARING



MACHINE TOOL WAYS

TABLE I—Properties of Plastic Bearing Materials

Property	Materials(1) Nylon	"Teflon"	Acetal
Coefficient of Friction (Dynamic) on Steel			
Dry	0.15-0.40	0.04-0.25	0.15-0.35
Water	0.14-0.19	0.04-0.08 (2)	0.1 -0.2
Oil	0.02-0.11	0.04-0.05 (2)	0.05-0.1
Fatigue Endurance Limit (psi)	3,000	1,400 (3)	5,000 (4) 3,000 (5)
Service Temperature (Max F)—			
Continuous Duty	150 (6)	500	185
Intermittent		600	250
Thermal Coefficient of Linear Expansion (in./in./°F)	4.6 to 5 x 10^{-5}	6.9 x 10^{-5} 3.6 x 10^{-5} (7) (77-212°F)	5.5 x 10^{-5}
PV Value (psi x fpm)— Continuous Duty-Max. (8)			
Dry	1,800-3,000	1,000 10,000 (7) 30,000 (9)	2,000-3,000
Water Lubrication	2,000-3,000	....	2,000-3,000
Oil, Initial	3,000-5,000	....	3,000-5,000
Oil, Wick	10,000-15,000	....	10,000-15,000
Water Absorption @ 72° F (per cent by weight)			
50 per cent R.H.	2-2.5	None	0.2
100 per cent R.H.	8.5-8.8	None	0.58
Submerged	8.5-8.93	None	0.9
Length Increase Caused by Moisture @ 72° F (per cent)			
50 per cent R.H.	0.5-0.6	None	0.05
100 per cent R.H.	1.8-2.5	None	0.2
Submerged	1.8-2.5	None	0.35
Taber Abrasion (mg after 1000 cycles from 1000 gram load and CS-17 wheel)	6-8	8 50 (7)	20
Yield point at 73 F, psi	8,500-14,600	850-1800 (7)	10,000

(1) Both nylon and acetal are members of comparatively broad families of polymers, properties of which may vary depending upon specific material selected, hence a range of values is sometimes shown. Nylons represented include "Cadco N-1"; "Zytel 101." Acetal is represented by "Delrin" 500x, produced by E.I. DuPont deNemours & Co. Values shown are intended as a theoretical design guide for bearing and material selection.

(2) 367 fpm and 5 lb. load.

(3) 7MM cycles.

(4) At 73° F and 100 per cent relative humidity.

(5) At 150° F and 100 per cent relative humidity.

(6) Heat stabilized nylons e.g. can be used to 275° F for continuous operation, 350° for intermittent operation.

(7) Filled compositions.

(8) Some designers select 75% of these ratings. However, prototype testing under operating conditions must determine actual feasibility.

(9) "Teflon" fiber—normally suggested for speeds up to 50 fpm.

Plastics . .

application of the criteria may be substantially different.

Of the three most widely used materials—nylon, TFE, acetal—each may have substantial advantages for any particular application. If non-lubricated, non-slip-stick operation is a prime consideration, the choice of materials would be TFE or acetal. If abrasion resistance is most important, the choice would be between nylon and Teflon. If high temperature operation is a prime requisite, Teflon might be best. If loadings are extremely high, the selection would be made between acetal and various filled TFE compositions.

Table No. 1 lists available properties of the three materials. Of particular interest in plane surface bearing design is the high yield of acetal. Although high for a plastics material, it nevertheless is quite low for metals. This suggests that virtually all plastic sliding bearings should be provided with a rigid backing, usually of metal. And although acetal is the most rigid, with a modulus of 410,000, nylon is a tougher material as shown by its slightly lower modulus and high yield point.

The Tabor abrasion tests also are of importance to designers concerned with plan surface applications, particularly because of the difficulty of enclosing or shielding surfaces in linear motion. As the chart shows, nylon has the greatest abrasion resistance, acetal the least. This suggests that in adverse conditions where shielding or shrouding is impossible, nylon would be the preferred plastic. In addition its ability to imbed (actually to absorb or engulf) foreign particles without significant effect on bearing properties further recommend nylon where abrasives are present. This imbedability characteristic also minimizes wear and scoring of the other surface. (Teflon also rates excellent in abrasion resistance.)

The static and dynamic coefficients of friction of both TFE and acetal are virtually the same. This eliminates stick-slip or jerky operation during starting or "inching-along," a major problem with conventional materials in plane surface applications.

Even initial lubrication will substantially improve the coefcients of friction for all three materials. In those linear bearing applications where grease may be applied, the improvement may be even more marked.

PV values, a commonly applied criteria in sleeve bearing design, also have applicability in plane surface bearings. PV refers to the product of the velocity in fpm and the bearing pressure in psi over the projected bearing area.

Application of these limiting factors (as shown in *Table II*) when designing plane surface bearings, is a much less precise or accurate guide than in sleeve bearing design.

Normally the problem of load in plane surface bearings may be more acute than in sleeve journal bearings, although distribution of load usually is over a large area. Speeds generally are lower. The major problem of PV limits therefore, becomes one primarily of control of frictional heat.

Deformation With Time

Cold flow is always present in plastic materials. This is a mechanism of yielding similar to that experienced with metals at high temperatures. For practical design purposes, additional deformation at less than one per cent strain after a one year period reaches a magnitude approximately equal to that of the initial strain. To estimate total deformation, the initial strain mav be multiplied by two.

The best way of allowing design latitude is by use of apparent modulus which is the initial modulus plus an allowance for long time deformation under load.

For the designer concerned with plane surface bearings, creep ca i cause problems. For instance f a metal bearing shoe or pad continually returns to rest at th : same spot on a plastic rail, the pad will, in time, deform the section of the rail beneath it more than the rest, in essence creating a valley. If the shoe must continue to pass back and forth over the valley it may create objectionable vibration.

In such an application, therefore it might be advisable to make the rail of metal, the shoe of plastic. Since total deformation is dependent upon thickness of material, deflection may be minimized by use of thin sections.

Of the three plastics under discussion, acetal is least affected by cold flow, TFE the most. The extremely high creep rate of pure TFE may be minimized by the use of filled materials which employ graphite, glass fibers or glass cloth as a strength supporting body.

Nylon, acetal and TFE may be used in maximum environmental temperature of 150, 185, and 500F respectively. These materials are poor conductors of heat. Hence it is advisable when applications are near maximum PV design limits, to provide some means for additional heat dissipation.

Normally it is best practice to use plastic for only one side of the bearing surface, with metal employed on the other to improve dissipation of heat generated bv friction.

Two other approaches are possible: making bearing sections as thin as practicable and backing them with metal, or by use of liquids. In plane surface bearing applications, the latter method may be least acceptable, although oils, water, grease, even liquids being processed

have been successfully used. The major problem, of course, is a method of retrieving and containing such liquids.

All three plastics have coefficients of thermal expansion several magnitudes greater than metals (see Table 1.) Thus, if temperature rise from ambient is substantial, adequate provision must be made for this expansion. Other than providing coolant, two approaches may be used.

1. Since total expansion is a dependent upon thickness or size of the material, it may be minimized by use of thin sections.
2. If this is undesirable and tolerances must be held extremely close, it is possible to design the mechanism to give precise operation at optimum top temperatures after a period of less precise operation during warmup. And since plastics are such poor heat conductors the warmup time is usually short.

In an incredibly wide number of applications, particularly in the field of materials handling, the problem of sliding friction is considerable. Unfortunately many such problems are incapable of solution by precise formula, since operating conditions may be extremely erratic and intermittent. These areas include buffer strips on conveyor chutes, rub rails, chute linings or coatings, etc.

As a general rule, nylon is probably the best material for rub rails and bumper strips, since the material is extremely abrasion resistant and has a low coefficient of friction. Its impact resistance and damping quality enable the material to absorb shock without damage.

Nylon has been used on such diverse items as ball grinder and piston feed chutes to eliminate scoring or nicking of the product, and to protect the chutes. ■

Some More Troubles

After several months of good operation, the second Scotch Yoke mechanism began to transmit erratic signals once again. A visual examination did not indicate any obvious troubles, so each sub-assembly was disconnected and checked for excessive looseness or broken parts with no success. The motor was operated with no load: the rotation of the output shaft was observed to be smooth and the sound of the speed reducer was quiet and steady. At this point, Aaron said, "Well – it's got me beat – let's take the whole unit back to the shop and we'll strip it down piece by peice to see if we can find anything."

An instrument-maker did just that and could find nothing wrong. "There's nothing left to suspect but the speed reducer in the motor-housing," said Aaron, "even though it seems to run smoothly and quietly." With this the instrument-maker picked up the motor and tried to turn the output shaft by hand -- and it did turn, making a rasping noise as it did so. "You've got a stripped gear in there, mister," he said. "That's good," Aaron said, "at least I'll know what needs fixing."

The speed reducer was quickly taken apart and every gear was looked at hopefully, but none of them had any teeth missing! At this point, Aaron decided to clean out all of the old grease so that he could look for scuff or scratch marks on gears, shafts or the gear case walls, and he did find one gear with scratch marks on the web. In order to determine why they were there he re-assembled the gear train and it was then that he noticed that one shaft and one of its supporters were worn in a peculiar manner (see photographs Exhibit B-1) which permitted two gears to move out of mesh. This turned out to be the cause of the latest trouble; the corrective action taken was to replace the motor/speed reducer combination with a new one of the same model – on the assumption that the observed wear had its beginning in an adverse combination of manufacturing tolerances which somehow had gotten by the motor manufacturer's Quality Control. The new motor/speed reducer failed in the same way – this time in a matter of a few weeks. During these few weeks, the original motor was returned to the manufacturer's engineering department for analysis (see Exhibit B-2).

With the failure of the second motor, the decision was made to redesign the drive to the Scotch Yoke. While this decision was being implemented the unit was driven by a third motor of the same kind as the original one. As of this writing it has been operating for six weeks with no sign of trouble.

EXHIBIT B-1

BODINE
ELECTRIC
COMPANY

2500 W. BRADLEY PLACE, CHICAGO, ILLINOIS 60618 AREA CODE 312-478-3515 TELEX 25-3646

ADDRESS REPLY TO:
J. F. CADY
DISTRICT REPRESENTATIVE
1485 BAYSHORE BOULEVARD
SAN FRANCISCO, CALIFORNIA 94124
PHONE: 415-467-8656

March 27, 1969

SLAC
P. O. Box 4349
Stanford, California 94305

Attention: Mr. A. Baumgarten - Engr.

Dear Mr. Baumgarten:

This will confirm our telephone conversation of March 26th.

The motor you sent back to Bodine-Chicago, Serial number 818RG046 was inspected and evaluated by our engineering department. We feel that the cause of failure can be pinpointed to two reasons:

#1. The gearmotor was defective to begin with....or

#2. At some time during the life of the gearmotor the load was considerably in excess of the mormal rated load.

In order to determine which of the above caused the motor to fail we suggest the following: We are repairing the motor at no-charge to Stanford, and returning it to you, making absolutely certain the motor is in first class condition before it is returned. You can then test the motor under full load conditions. Should it fail again, we will be certain that the motor is being misapplied, (from a torque standpoint). In this case our suggestion would be to switch to the N-1RD gearmotor.

From previous tests and experience with this type of motor, it would appear that this is a case of simply too much load for this motor. While the gears may stand up under these conditions, the studs will not. It should be noted that this motor was subjected to over 2000 hours of operation in it's twelve weeks of use. If more than 2000 hours is desirable, a larger stronger motor should be used to give you a service factor, proportional to the expected life you desire.

BODINE
ELECTRIC
COMPANY

2500 W. BRADLEY PLACE, CHICAGO, ILLINOIS 60618 AREA CODE 312-478-3515 TELEX 25-3646

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DISTRICT REPRESENTATIVE
1485 BAYSHORE BOULEVARD
SAN FRANCISCO, CALIFORNIA 94124
PHONE: 415-467-8656

March 27, 1969

For example: Motors and gearmotors are designed to give the user between 2500 and 5000 hours of operation under nameplate conditions. You'll find some that are designed to give more than this, but they have to be placed in the category of special service motors, not general service as your present motor is. So, if you have a requirement for 10,000 hours you should base your selection on a motor with a rating of four or five times your load under normal conditions. Normal being defined as eight hours a day, five days a week.

One point of caution. On any application requiring more than 5000 hours of operation, the manufacturer should be consulted as to instructions on general maintenance of that motor. Items to be watched very closely would be bearings and lubricants, and in the case of DC motors, brushes and commutator wear.

We hope that you will find this information helpful. If we can be of any further service to you or the Students of Stanford Univ. please feel free to call on us at your convenience.

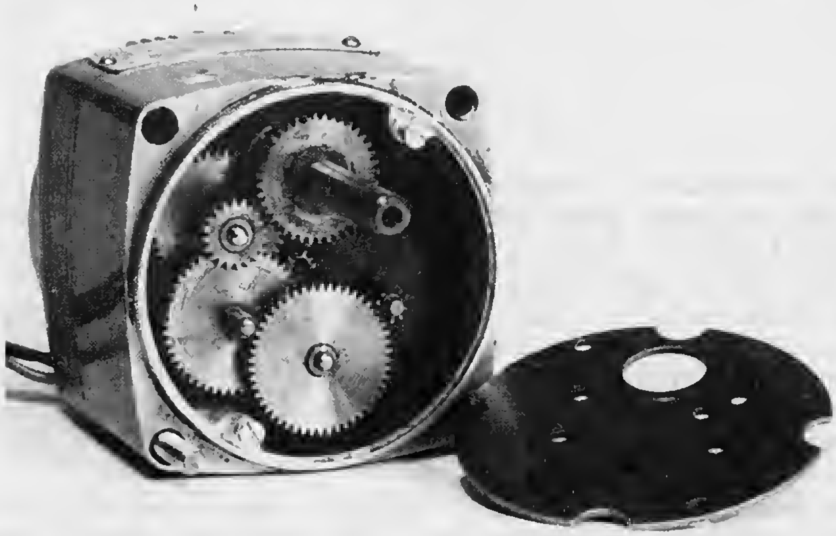
Cordially,

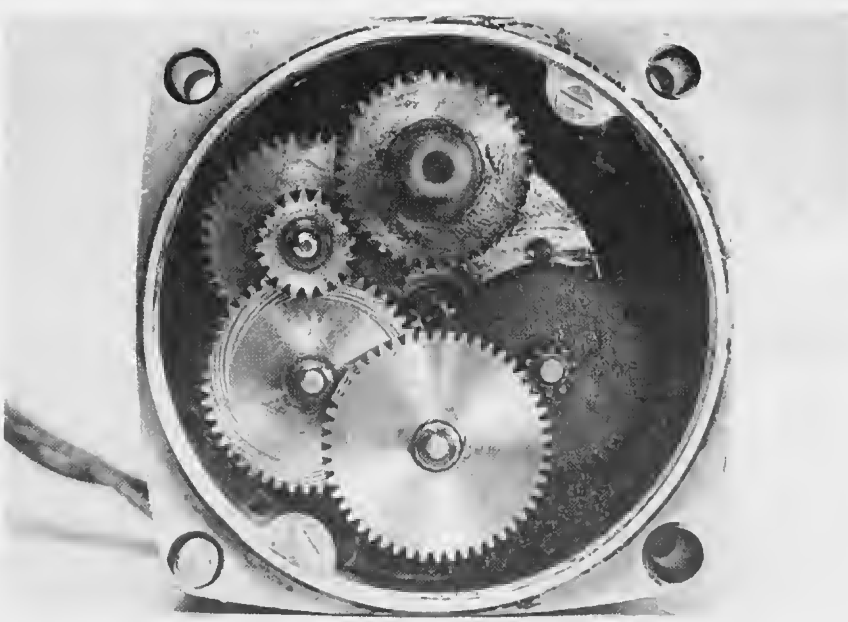
R. F. Cady

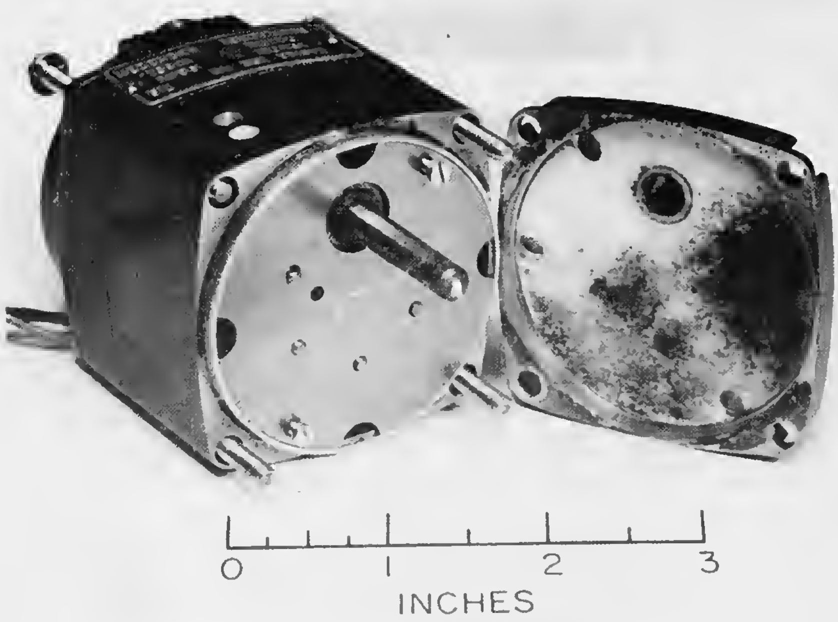
J. F. Cady Company
1485 Bayshore Blvd.
San Francisco, California 94124

EXHIBIT B-2

FAILED ELECTRIC MOTOR



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1
2
3
INCHES

INSTRUCTOR'S NOTES

This case shows, not perfection, but an actual design job, performed under constraints of time and money, for a single item. The choice of the Scotch Yoke over the simple eccentric cam or the slider crank was a decision based on the author's personal experience. The merits and shortcomings of these mechanisms, and the rational basis for engineering decisions, are fertile areas of discussion.

The particular configuration of the Scotch Yoke chosen by the designer is not the only possible configuration—and perhaps not the "best" from the viewpoint of all engineers. The differences can be examined in the light of design criterea which students would have to develop.

The suggested detail drawing assignments may be useful exercises in deffinition of shapes and dimensions. Suggested assignments No. 4 and No. 5 are exercises in the application of beam deflection theory.

SUGGESTED ASSIGNMENTS for part A

1. Review the various mechanisms shown in Fig. 2. Compare them on the basis of required number of parts, precision required of the parts, ease of manufacture of the parts, etc.

2. Sketch three possible different configurations of the Scotch Yoke. Strive for simplicity of manufacture, accuracy of performance, direct transmission of forces with the least possible bending, twisting, or binding.

3. Assume that you as a detail designer have been asked to draw up the first yoke (part 2 of Exhibit 2a), the second yoke (part 2 of Exhibit 6a), or the frame (part 1 of Exhibit 6a). Estimate the time you will require to do this job; record the time actually used. Use the sample detail drawings shown in Exhibits 2b, 2c, 6b, 6c, as models.

4. Compare the stiffness of the second design to that of the first design, on the basis of calculated deflections under unit force applied upwards or downwards by the crank at the positions of upwards and downwards motion of the crank.

5. Compare up-and-down motion of the yoke, caused by its weight acting at different positions of the supporting rods, for the first and second designs of the Scotch Yoke mechanism. Assume that the calculation of deflection shown in Appendix A refers to the center position of the yoke, and that the support rods in the later design are 3/8" diameter.

SUGGESTED QUESTIONS for part B

6. Estimate the highest torque imposed on the output gear by the operation of the Scotch Yoke.

7. Will the torque on the output gear reverse driving a cycle of operation?

8. What should the designer do now?

(Another part of this case, showing Dr. Banngarten's next step, is in preparation)